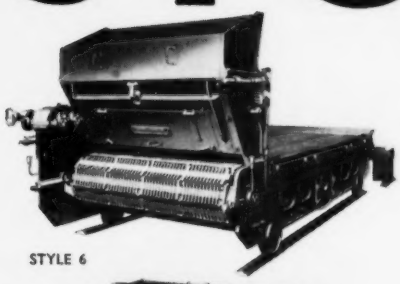
STOKERS



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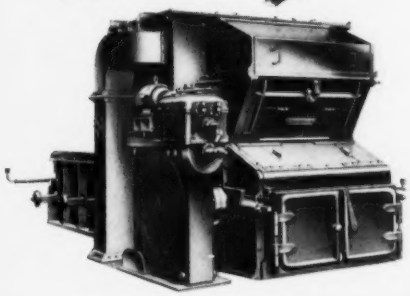

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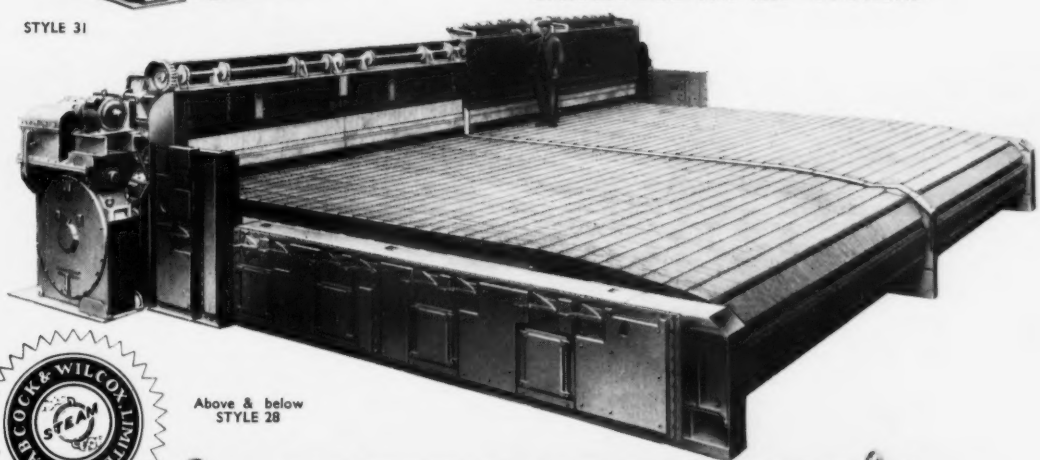
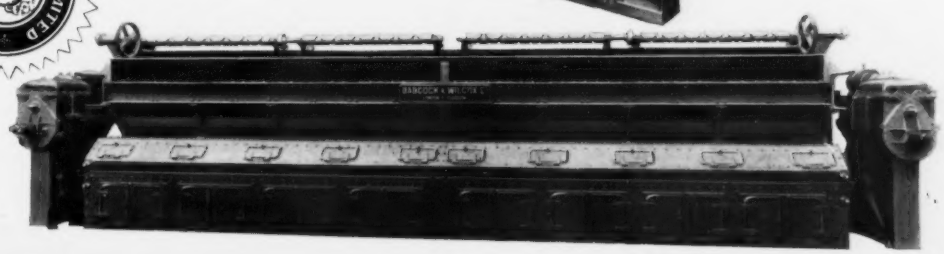


Above & below
STYLE 28



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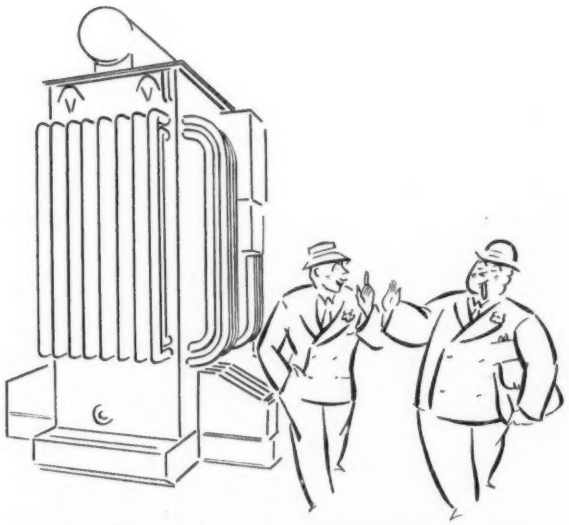

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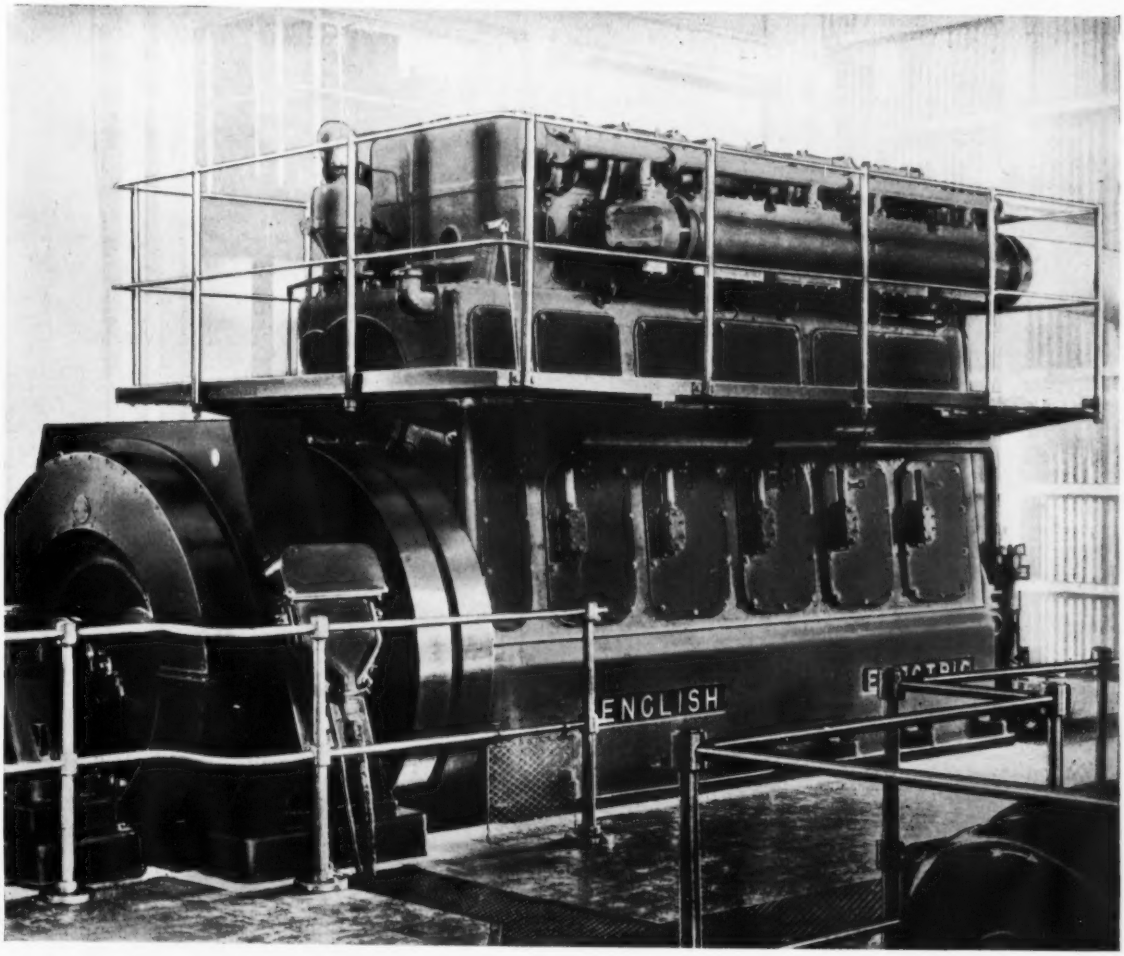

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† Bolton's made many important components for the "Bristol" aeroplane in which a world's altitude record of nearly 50,000 feet was established in 1936.

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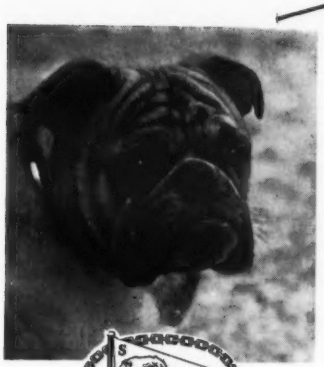

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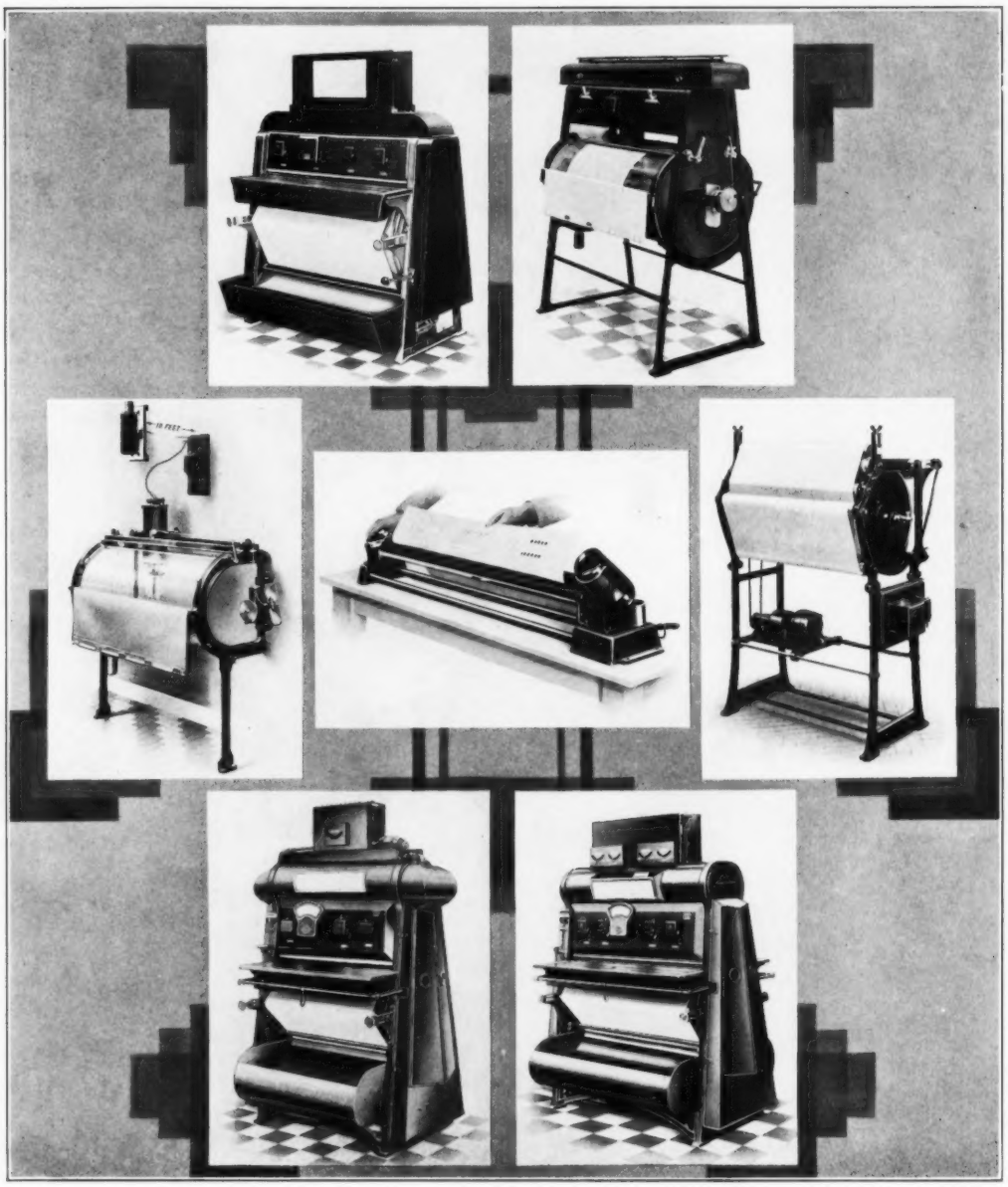

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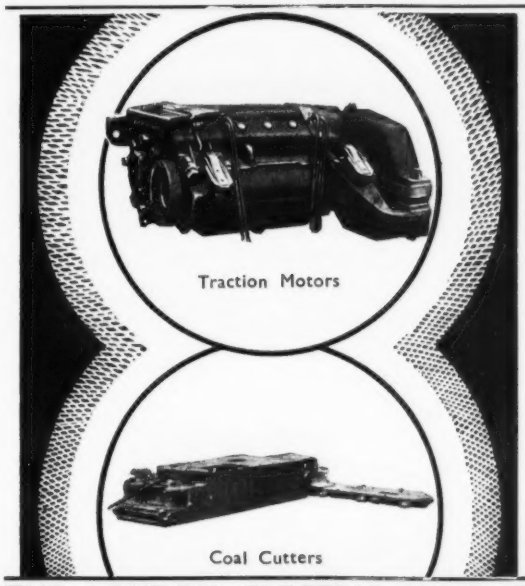

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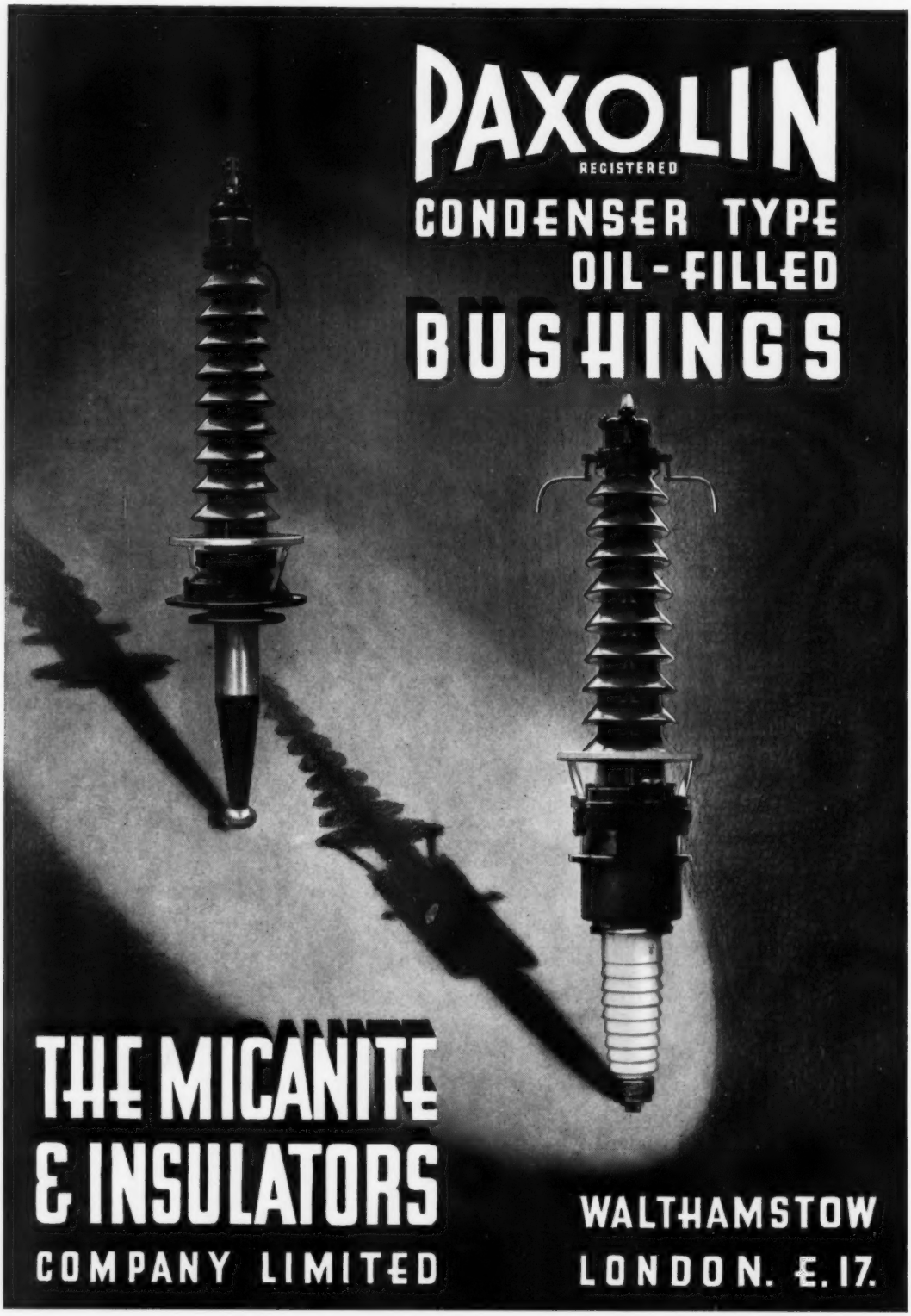
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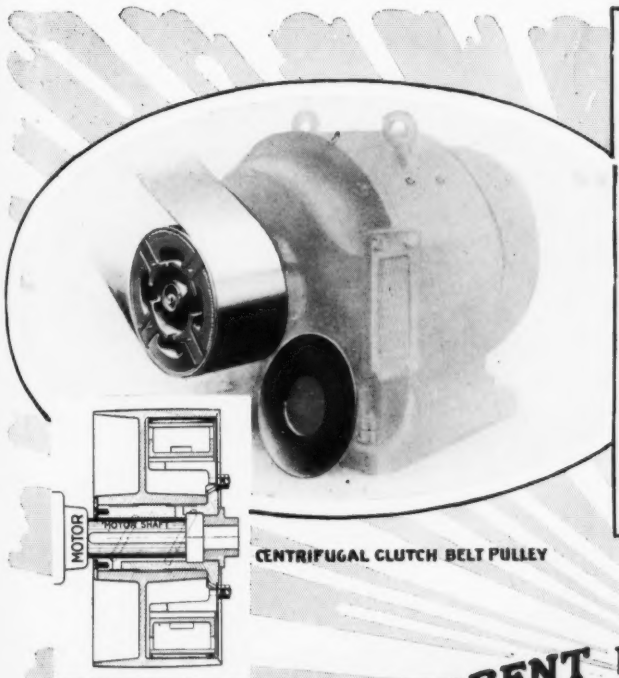

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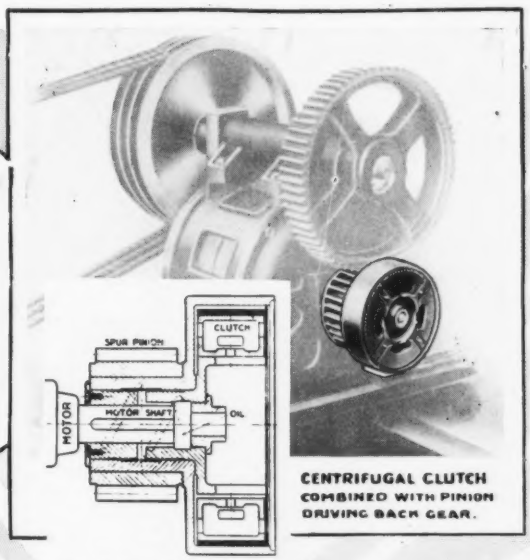

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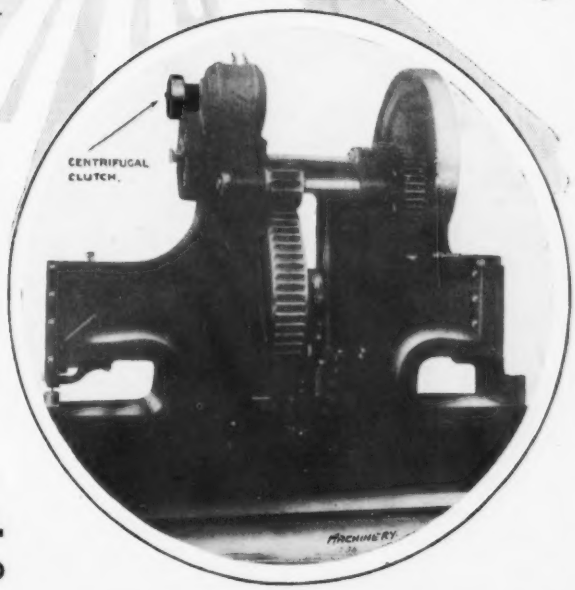

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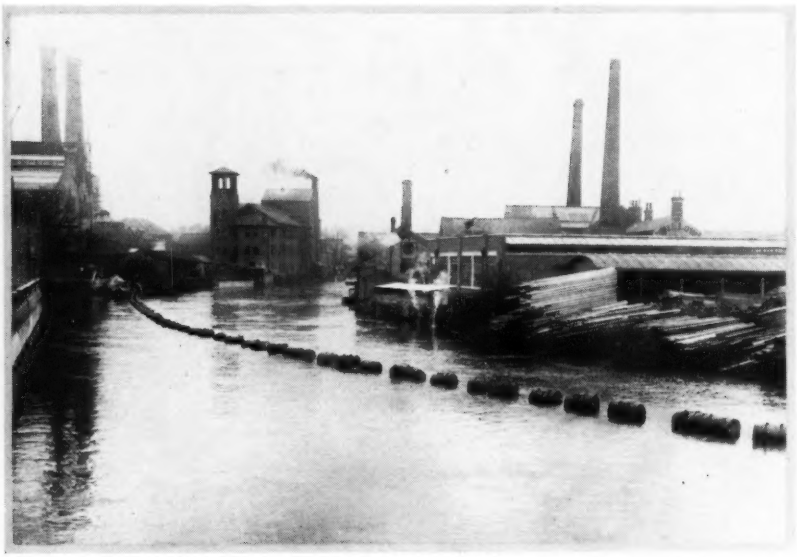

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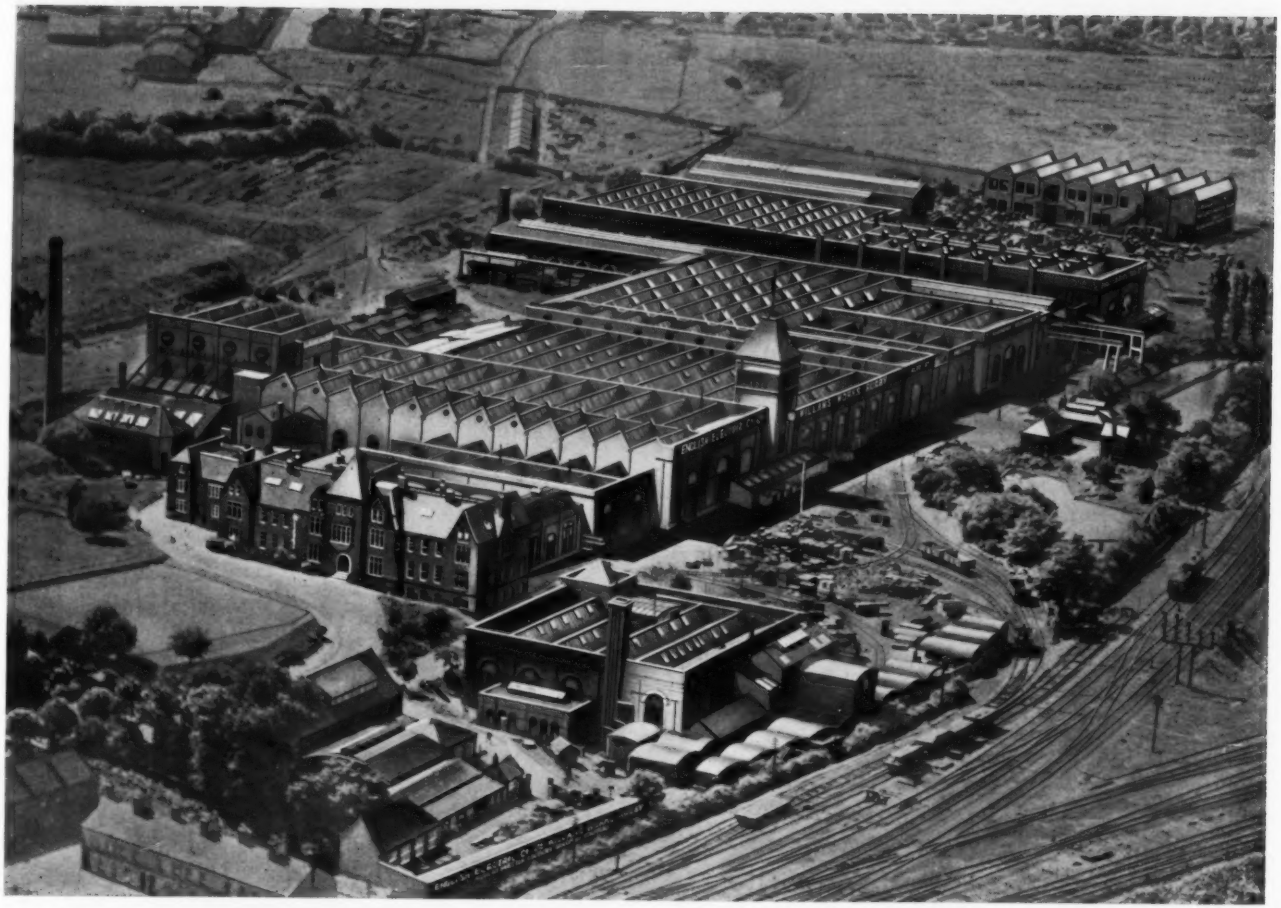


Aerial view of the English Electric Company's Rugby Works.

The English Electric Journal

Vol. VIII. No. 6 JUNE, 1937 One Shilling.

CONTENTS.

PAGE

RECENT DEVELOPMENTS IN HYDRO-ELECTRIC ENGINEERING WITH SPECIAL REFERENCE TO BRITISH PRACTICE - - 153

STEAM TURBINE RESEARCH AT RUGBY WORKS - - - 190

All Communications respecting Editorial Contents should be addressed to THE EDITOR, THE ENGLISH ELECTRIC JOURNAL, THE ENGLISH ELECTRIC COMPANY LIMITED, STAFFORD.

EDITORIAL.

The Institution of Mechanical Engineers is this year holding its Summer Meeting in the Midlands, and during the course of the meeting the members will visit our Rugby Works, where our steam turbines, water turbines and diesel engines are manufactured.

In the circumstances we consider it appropriate to devote this issue of the Journal to our Rugby products, not in the form of articles descriptive of those products, but of articles dealing with certain new developments and experimental work. We hope that these will be found to be of more than usual interest to all our readers.

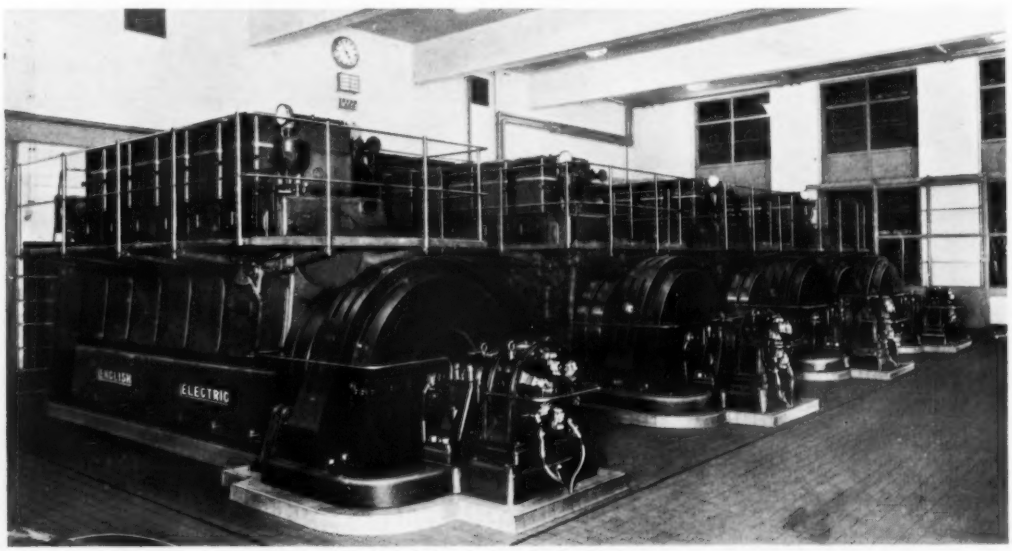
First we give in abstract a paper read recently by Mr. P. W. Seewer, the Company's Chief Hydraulic Engineer, before the Institution which obtained for him the Willans Premium of the Institution granted for the best paper dealing with the utilisation of energy during the years 1931 to 1936.

The Author points out that in order to secure a share of the world's output of

hydro-electric machinery it was necessary for British manufacturers to create new features of design affording real advantages to prospective users. These new features form the main subject of the paper. The Company is the only British manufacturer of complete water turbine sets, comprising both turbine and electric generator, and, further, it is, we believe, the only Company that has a Hydraulic Test Station of its type in the country.

The remainder of the Journal is devoted to an article by Mr. J. P. Chittenden, the Chief Engineer and Manager of our Steam Turbine Department, which gives a review of work carried out by the Research Section of the Steam Turbine Department. The Author, after referring to pioneer work that has been carried out by the Rugby Works, proceeds to deal with the experimental turbine used in the development of new designs, together with the problems it has been possible to investigate by its use. He further deals with research on mechanical stability and vibration problems, and gives many illustrations of experiments that he has carried out.

Space has not permitted the inclusion of an article dealing with diesel engines manufactured at Rugby. These have frequently been referred to in previous issues of the Journal, and it must here suffice to mention that the Works have been making these engines for over thirty years, that they vary in size from 150 B.H.P. to over 3 000 B.H.P. and that last year the Company built an engine, of the Fullagar type, of 3,500 B.H.P., which is the largest diesel engine manufactured in Great Britain for industrial purposes.



Four 750 B.H.P. 6-cylinder "English Electric" "L" type Diesel Engines and Alternators at the Droitwich Station of the British Broadcasting Corporation.

Recent Developments in Hydro-Electric Engineering with Special Reference to British Practice.

Abstract of a paper read before the Institution of Mechanical Engineers, London, 18th December, 1936.

By P. W. SEEWER, D.E., M.Inst.C.E, M.I.Mech.E., Chief Engineer, Hydraulic Department.

Introduction.

Nearly twenty years ago our Company decided to extent its activities into the field of hydro-electric engineering. Our Works had, prior to this, designed and manufactured numerous water turbine driven generators, together with transformers, switchgear, and other electrical equipment appertaining to hydro-electric plants, and in 1920 it was realised that the inclusion of water turbines, together with other allied hydraulic equipment, would enable our Company to undertake the manufacture of hydro-electric plant on a comprehensive basis, and that this would be a desirable and fitting field for British industry.

In order to build up a sound business in hydro-electric plant, it was necessary :—

(*a*) To design and manufacture complete hydro-electric units in our own Works, so as to secure the great advantages of combined manufacture of turbine and generator, which are integral parts of one and the same whole.

(*b*) To possess a staff competent to deal with comprehensive hydro-electric schemes, and capable of devising and introducing new features of design and manufacture that would afford real advantages to the prospective user.

(*c*) To install equipment for testing our own water turbines, valves, and other hydraulic accessories, in addition to the equipment for testing electrical plant that we already possessed.

(*d*) To maintain the tradition of high-class workmanship and reliability enjoyed by British machinery.

The Company accordingly established an active Water Turbine Department, and a Hydraulic Testing Station, and the result has been that, to the present date, the Company has designed and manufactured hydro-electric plant of many hundred thousand horse-power capacity.

We give on the following pages an abstract of a paper read before the Institution of Mechanical Engineers by Mr. P. W. Seewer, Chief Hydraulic Engineer of the Company. In this paper are described several special features of design and construction, which can be briefly summarised as follows :—

(1) Automatic aerofoil type permanent water flow recorder.
(2) Automatic governor actuator and mechanical drive.
(3) Cylindrical balanced valve and discharge regulator.
(4) Hydro brake.
(5) Impulse wheel jet diffuser type governing system.
(6) Impulse wheel bucket fastening.
(7) Experimental and research plant with test results.
(8) General design and special mechanical features in British practice.

Extent of Water Power.

No systematic and exhaustive survey of the water power at the disposal of man has yet been attempted. The calculations of a number of geological and hydrographic surveys, however, give a practical idea of the water power available over the whole world, as compared with that actually utilized up to the present time. The figures in Table 1 can be taken as the power available at ordinary minimum flow and the considerable margin due to storage which is possible in a large percentage of catchment areas is omitted altogether. The figures are, therefore, conservative.

The total available energy will be seen to be about 500,000,000 h.p., without, of course, including any contribution derived from tidal power.

Table 1. Available Water Power.

	Probable energy disposable, h.p.	Utilized energy, h.p.	Utilized energy, per cent.
Europe	58,000,000	19,500,000	33.7
America (all) ...	127,000,000	33,000,000	26.0
North and Central America	73,000,000	30,000,000	42.0
Asia	90,000,000	5,400,000	6.0
Australia	17,000,000	850,000	5.0
Africa	190,000,000	380,000	0.2

Plate I gives a fair idea of the situation and distribution of water power over the whole world. The extent of utilization of water power in the British Empire is, apart from the Dominion of Canada, as yet very small compared with countries such as Sweden, Switzerland, Italy, Germany, and France, where cheap water power is depended upon as a factor of first importance in the national economy. The importance of water power within the British Empire was recognised by the British Parliamentary Water Power Committee, which submitted the following conclusions :—

(1) The potential water power in the Empire amounts in the aggregate to at least 60,000,000 to 70,000,000 h.p.

(2) Much of this is capable of sound economic development. (It may be added that about 20 per cent. of these schemes only could be classified as " high-pressure " schemes with 800-6,000 feet head ; 40 per cent. as " medium-pressure " schemes with 150-700 feet head ; about 20 per cent. as " low-pressure " schemes with about 150 feet head ; and the remainder as low-head schemes, which are the least economic and require extensive machinery and buildings for comparatively small powers.)

(3) Except in Canada and New Zealand, and to a lesser extent in New South Wales and Tasmania, no systematic estimate has yet been made by any Government Department to ascertain the true resources of its territories.

(4) The development of the Empire's natural resources is inseparably connected with that of its water powers.

To illustrate the importance of the utilization of water power, a diagram is given in Fig. 1, prepared by the Dominion Water Power and Hydrometric Bureau of Canada, showing the growth of hydraulic developments in Canada from 1900 to 1936.

In order to secure a reasonable share of the world's output of hydro-electric machinery, it was necessary for British engineers to create features in design and manufacture affording real advan-

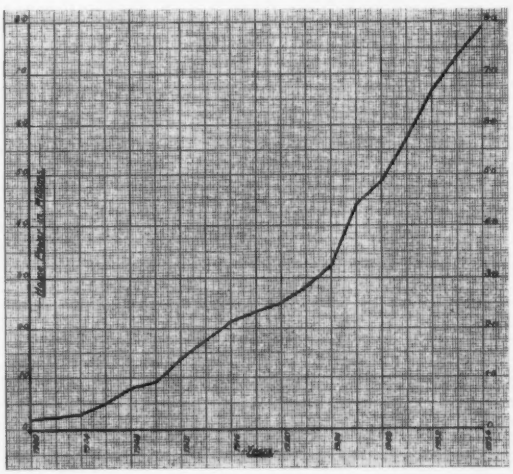


Fig. 1.—Chart of water power development in Canada.

180° 165° 150° 135° 120° 105° 90° 75° 60°
ARCTIC OCEAN
FRANKLIN
ALASKA
YUKON
MACKENZIE
27 MILLION HP
KEEWATIN
HUDSON BAY
BERING SEA
CANADA
BRITISH COLUMBIA
ALBERTA
SASKATCHEWAN
MANITOBA
ONTARIO
QUEBEC
LABRADOR
NEWF
60°
45°
U.S.A.
38 MILLION HP
Bermuda I.
30°
GULF OF MEXICO
MEXICO
8 MILLION HP
THE WEST INDIES
CUBA
HONDURAS
CARIBBEAN SEA
GUATEMALA
SALVADOR
NICARAGUA
COSTA RICA
PANAMA
SANDWICH Is
15°
PACIFIC
VENEZUELA
COLOMBIA
BRITISH GUIANA
Christmas I.
0°
ECUADOR
PERU
Marquesas Is
Samoa Is
Low Archipeligo
54 MILLION H
BOLIVIA
BRAZ
Paraguay
Cook Is
OCEAN
Uruguay
CHILE
ARGENTINA
Chatham Is
45°
Falkland
Cape Horn
180 165° 150° 135° 120° 105° 90° 75° 60°

ER POWER OF THE WORLD.

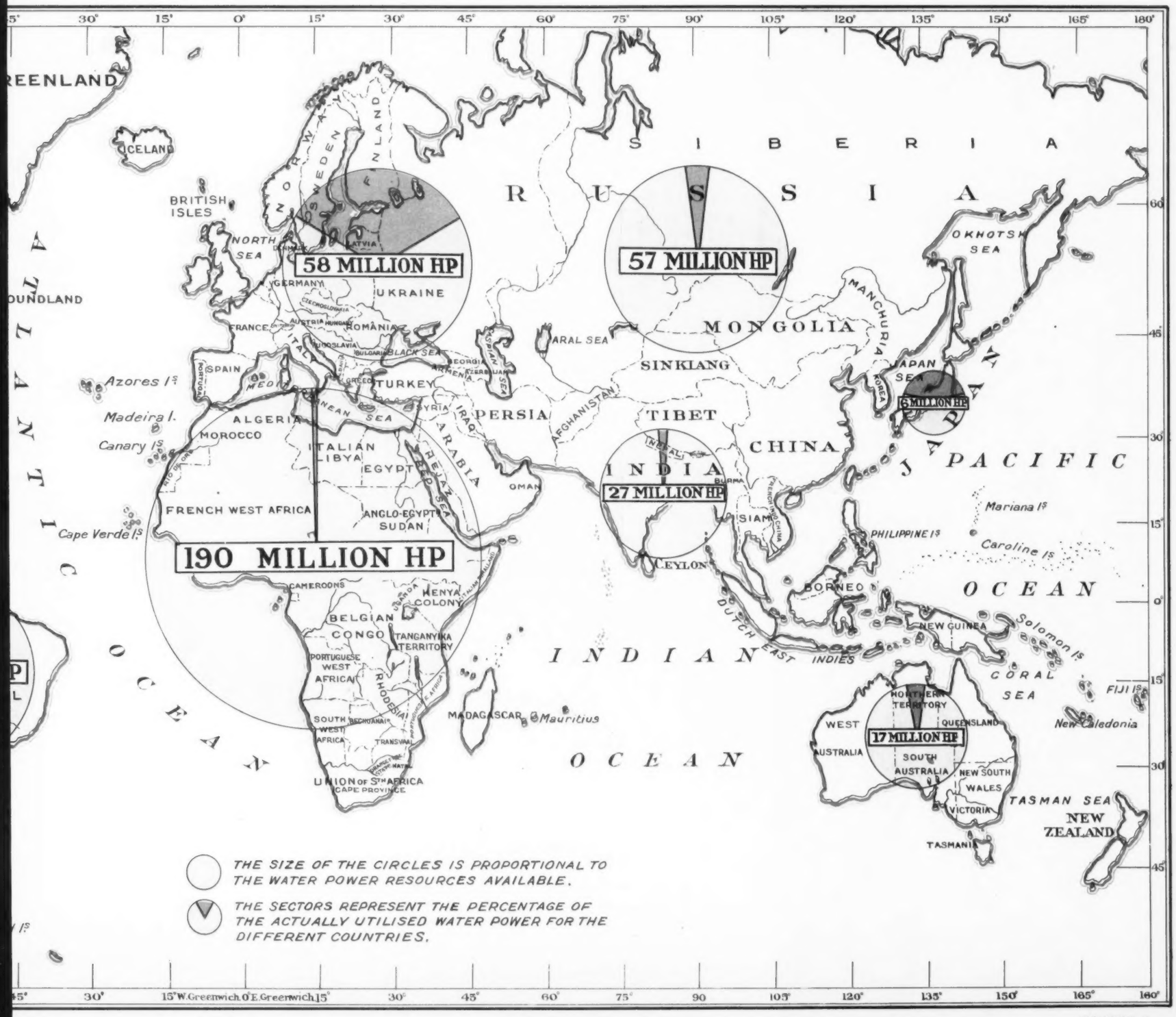


PLATE I.

tages, both technical and economic, to the prospective user. It is the purpose of the paper to describe briefly some of the more important of these features which have met with approval, especially amongst water power engineers of considerable experience. The main requirements were simplicity, high specific speeds for both reaction and impulse wheels, exact and automatic regulation for the turbines proper, and reliable control apparatus such as governors, valves, and discharge regulators.

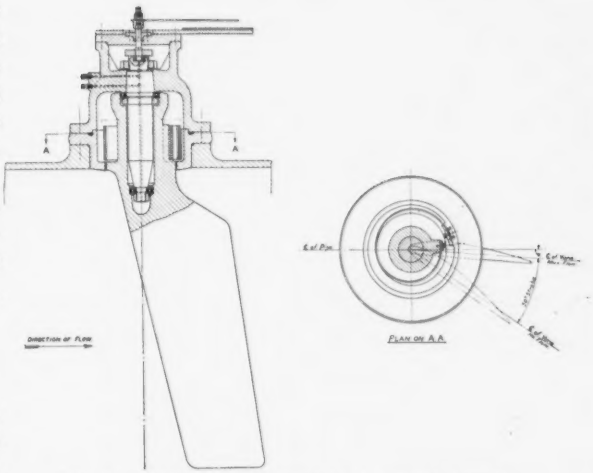


Fig. 2.—Sectional arrangement of Automatic Aerofoil Flow Recorder.

Automatic Aerofoil Flow Recorder.

Hydro-electric power plants involve large financial investment and therefore demand accurate knowledge of all operating factors, and continuous supervision by means of records of power input and output. Electrical metering and recording instruments capable of a high degree of accuracy are available. The exact measurement and continuous recording of the rate of flow of large quantities of water in the field, however, is still a most difficult problem. Exact and continuous flow recording is of paramount importance for the maintenance of high efficiencies, and for the control of storage reservoirs and concessional water quantities, including irrigation flow. As the output, head, speed, and other hydraulic conditions are continuously changing in service, it is desirable to install apparatus capable of recording accurately the quantities of water discharged in a permanent manner, and independent of all the other variables.

The well-known methods of water measurement by Pitot tube, Venturi meter, salt velocity (Professor Allen), friction loss calibration, time-pressure (Gibson), differential pressure recording by piezometers on the intake vortex, or by weir calibration, while mostly capable of yielding accurate and consistent records indicating the *momentary* rate of flow, are mostly unsuitable or quite impracticable for *continuous* flow recording. Venturi tubes for plants of medium or low heads having pipes of large diameter extending sometimes to 20 or even 30 feet, are cumbersome and expensive, impose appreciable pressure losses, and influence detrimentally the regulating conditions. In common with Pitot tubes, they are subject to clogging by dead leaves, sand, and other detritus carried in considerable quantities by many rivers. Furthermore, none of the orthodox methods of water flow recording afford any appreciable power which can be used reliably for the operation of bypass valves or other control devices.

These considerations led to the development of the aerofoil type of permanent water flow recorder shown in section in Fig. 2 and illustrated in Fig. 3. Aeronautical principles have established that a suitably designed aerofoil in a stream of air sustains forces, notably an uplift, which are direct functions of the shape of the aerofoil and the angle of incidence. Fig. 4 shows a characteristic aerofoil curve, in the form of a polar diagram giving the uplift and resisting forces, the moments established in relation to the front edge, and the angles of incidence. It was also established by theory and experiment that an aerofoil immersed in flowing water, for instance inside a closed pipe, and mounted so as to be capable of rotation, was highly sensitive to changes in the velocity of flow and would take up a definite position for any particular velocity. The torsional moments exerted by the flowing water are balanced by a simple coil spring and the aerofoil is mounted in two

Fig. 3.—Automatic Aerofoil Flow Recorder.

rustless steel ball races permanently working in grease, while an extension of the pivoting shaft operates the flow indicator, recording apparatus and any control devices. The aerofoil can be used for the operation and adjustment of relief valves or other discharge apparatus in order to establish a permanent or variable flow according to requirements. This is especially useful in cases where, as for irrigation purposes or for superimposed plants along a river, an exactly predetermined flow must be kept to and guaranteed, independent of the momentary load, which, however, must be adjustable according to the constantly varying requirements. The aerofoil is a very simple and robust apparatus which is independent of the pressure prevailing in the pipe or duct and is free from disturbing influences due to impurities contained in the water. There is no discrepancy between the flow as measured by a weir and as indicated by the aerofoil meter, except below 10 per cent. of maximum flow, where the degree of inaccuracy of the latter is of the order of $\frac{1}{2}$ per cent. Fig. 5 shows the operating characteristics of an aerofoil flow meter in relation to a calibrated weir. For convenience the experimental results were superimposed on a mathematically deter-

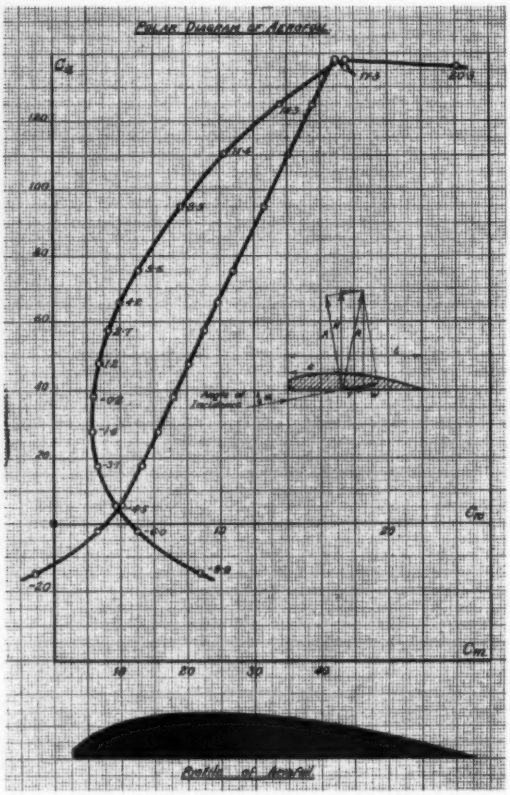


Fig. 4.—Polar Diagram of Aerofoil.

$$C_a = \frac{A}{qF} \times 100\,;\quad C_w = \frac{M}{qF_l} \times 100\,;\quad C_u = \frac{W}{qF} \times 100$$

$q = \frac{1}{2}\rho v^2$; $\rho = \gamma/g$ = specific mass of fluid; g = gravitation constant; γ = specific weight of fluid; F = area of aerofoil = width × length; A = lift perpendicular to flow; W = drag in direction of flow; R = resultant of W and A; M = momentum of resultant R around inlet edge.

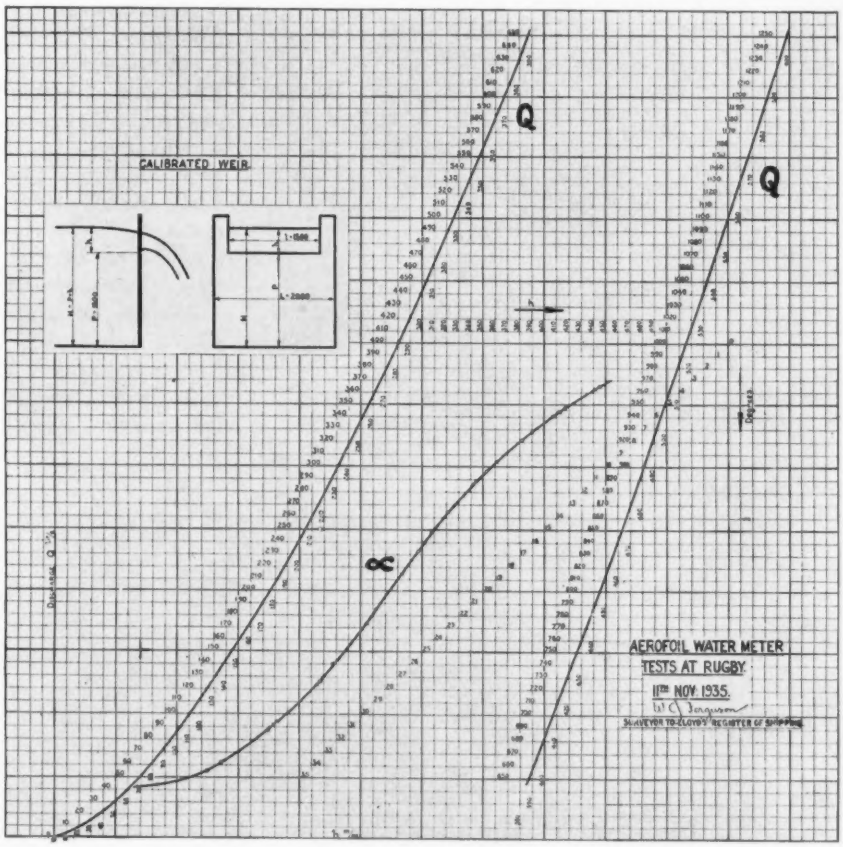


Fig. 5.—Quantity Chart of Automatic Aerofoil Flow Recorder in relation to Calibrated Weir. α = degrees of Aerofoil Readings plotted against head on Weir h. Q = Quantity plotted against head on Weir h.

mined calibration curve for the weir.* Fig. 6 gives an example of a recording chart obtained in the field, and Fig. 7 a curve of moments, which shows how powerful the instrument can be. The aerofoil is free from vibration in service and no measurable detrimental influence whatever on the efficiency of the turbines can be detected.

The Automatic Governor-Actuator and Mechanical Drive.

The automatic governor-actuator can aptly be called the brain of the whole plant, as it controls the speed, load, and pressure variations. Upon it depends the safety and uninterrupted service of the plant. A governor-actuator must ensure : maximum safety under all conditions of service ; high sensitiveness with freedom from vibration and " hunting " ; compactness and freedom from wear, that is, lost motion, and it must be fool-proof ; easy accessibility, sturdiness, and simplicity of every part ; and standardization so as to be adaptable to every kind of service, and every size of plant. The " English Electric " automatic governor-actuator shown in section in Fig. 8 is made up of three main parts :—

(1) The centrifugal device or governor head.

* Tests conducted and witnessed by Lloyd's Register of Shipping.

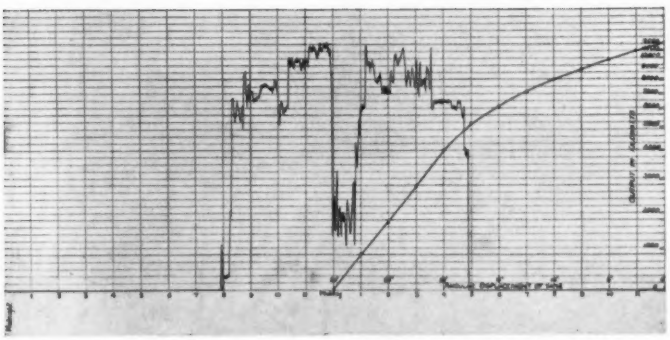


Fig. 6.—Aerofoil Recorder Chart taken at the Glenlee Power Station of the Galloway Water Power Company.

(2) The actuator gear, with which is combined the double-acting return motion gear or compensating device.

(3) The oil pressure relay and main steering valve through which the oil pressure is distributed to the servo-motor.

The governor also comprises a number of accessory parts which permit synchronization through speed adjustment by hand and remote control, several safety devices in case of oil pressure or drive failure, the necessary starting and stopping devices, both hand-operated and automatic, and a dial tachometer.

The centrifugal device is of the hydraulic type and comprises two compensated flyweights in cam form, which control the differential pressure on one side of an oil pressure relay by regulating the oil discharge through two nozzles connected with this differential pressure chamber, the piston of which revolves with the flyweights. The latter are suspended by two springs and linked together by two coil springs by which the centrifugal forces developed in rotation by the flyweights are counterbalanced. The centrifugal device contains neither fulcrum pins nor knife-edges, and its relay piston moves axially while rotating and while completely immersed in oil. The flyweight cams regulate the position of the relay piston by controlling a thin oil film at the nozzles. The device contains no parts subject to wear and is extremely sensitive. It rotates in two ball races which have no influence on its sensitiveness.

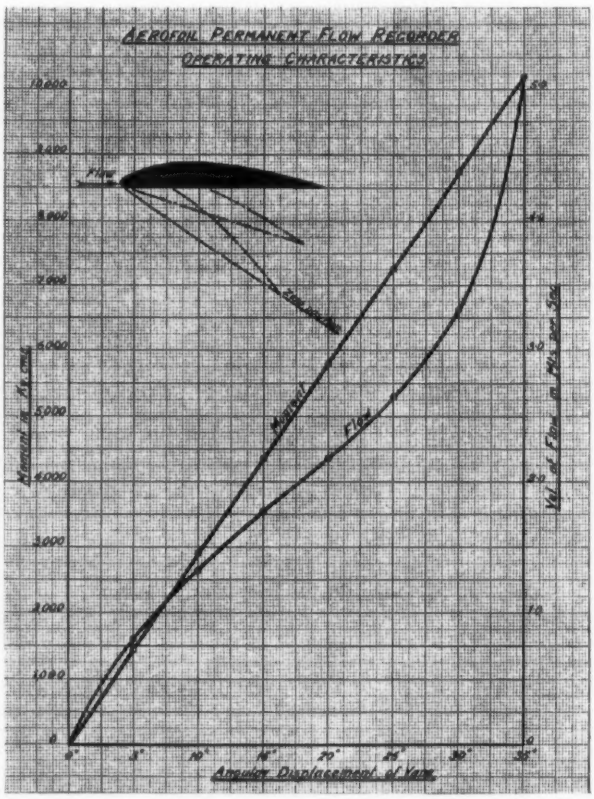


Fig. 7.—Flow Momentum Chart.

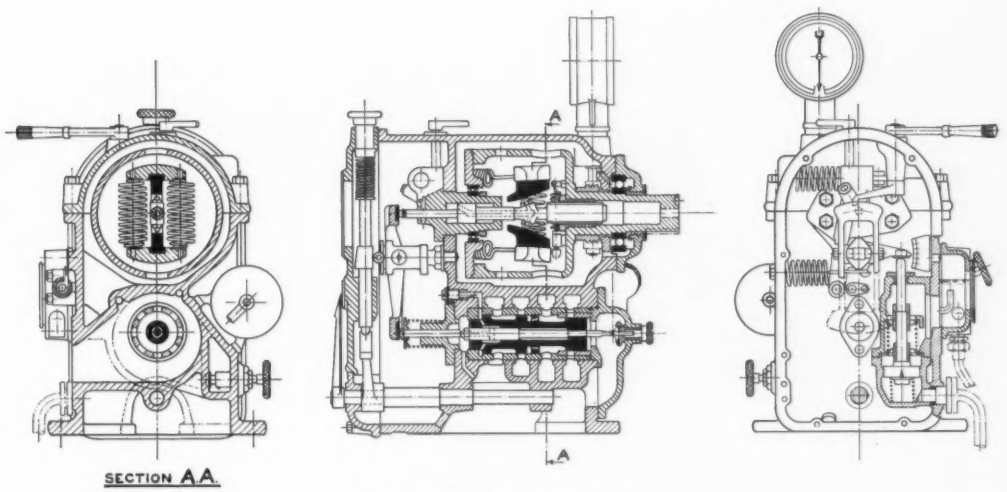


Fig. 8.—Automatic Governor Actuator.

The actuator gear and compensating device are concentrated on one single pivoting lever, which has a treble movement as follows. It transmits the movement of the rotating pendulum pin to the operating valve underneath by pivoting, thus causing the relay of the main operating or steering valve to move in accordance with the speed. A small arm on the sleeve which supports the actuator bridge is in continuous contact with the hydraulic return motion or compensating device connected with the servo-motor. The latter neutralizes the steering valve after the initial operating movement by means of inclined compensating rails mounted on the actuator bridge, this being the secondary pivoting movement. The compensating rails are also pivotally adjustable so that the temporary statism or damping can be adjusted to suit any plant conditions. The third and translatory movement is imparted to the actuator bridge by the second and mechanical return motion device by means of a cam and roller, which brings about the permanent speed

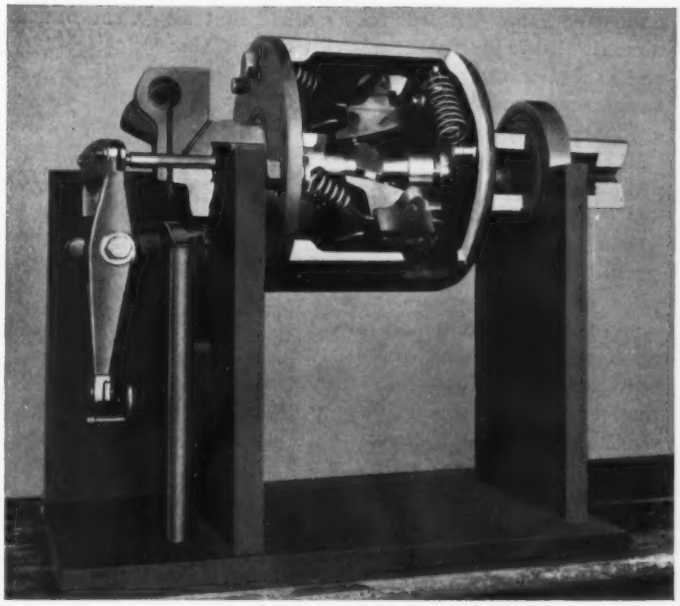


Fig. 9.—Model of Section of Governor Pendulum.

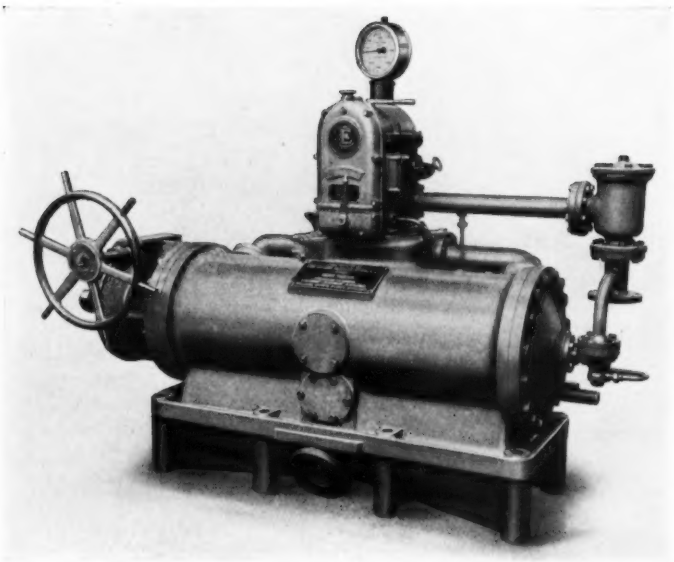


Fig. 10.—Governor Actuator and Servo-Motor.

variation or permanent statism of the governor. This is adjustable by hand by a simple cam in continuous contact with a roller on the actuator bridge mounting. It will be remarked that all contacts always bear in the same direction so that no lost motion can develop without immediately being compensated. This is important for the elimination of " hunting."

The return motion or compensating device is of the hydraulic type and operates by means of a compensated and spring-loaded neutralized dash-pot connected by pipes with the servo-motor piston. It cannot develop lost motion and its operation is asymptotic in both the closing and opening directions. It makes it possible to operate the turbine with the same regularity and steadiness for any degree of statism down to zero when there is no difference in speed between full load and no load (" isodrom " regulation).

A flexible mechanical governor drive, Fig. 12, has been adopted by the English Electric Company as standard on all important sets. It permits of direct mechanical drive with its attendant reliability and yet embodies the necessary elasticity to eliminate vibration and the detrimental effect of temperature changes. It consists of a spur and bevel-gear drive connecting the governor head to the turbine shaft by a shaft into which a flexible and oval-slotted stainless steel plate is inserted. The latter provides the required combination of rigidity and flexibility. The gear drive operates entirely in an oil bath, the oil being constantly renewed by the oiling system of the bearings. Fig. 13 shows some typical tachogram curves of the " English Electric " governor which are very consistent.

Cylindrical Balanced Valve and Discharge Regulator.

The design and manufacture of comprehensive hydro-electric plants showed the necessity of

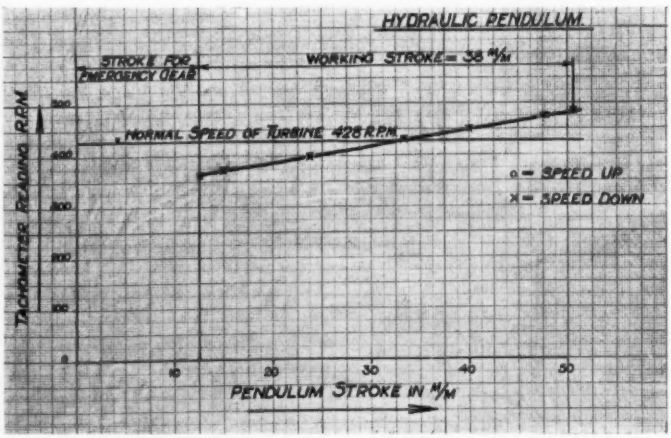


Fig. 11.—Pendulum operating characteristic obtained by means of Electric Speed Adjustment, Micrometer measurements, and Dial Tachometer control.

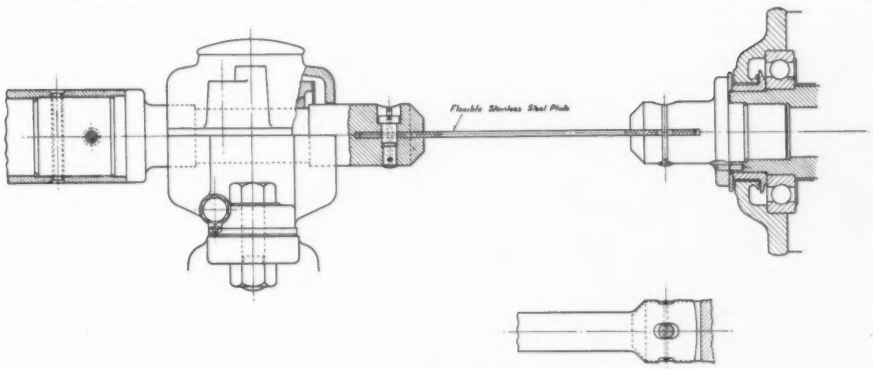


Fig. 12.—Flexible Mechanical Governor Drive.

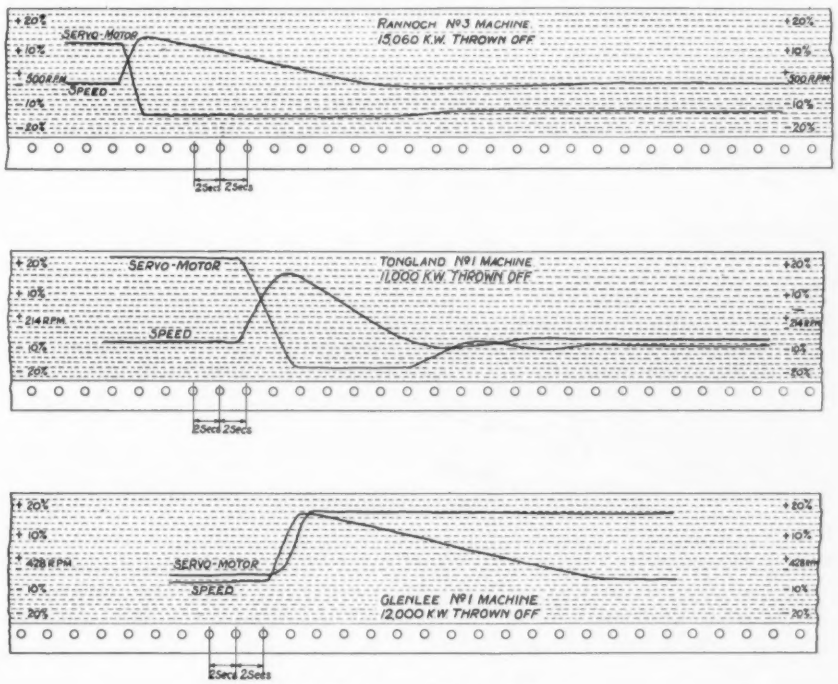


Fig. 13.—Tachogram Curves of Governor.

designing stop valves and relief valves, that is, discharge regulators of the Company's own type which could be embodied conveniently with their governors and safety gears and could be relied upon to operate safely under the most exacting conditions.

Discharge regulators must ensure rapid and smooth closing and opening combined with absolute safety and freedom from water hammer. The valve, as illustrated in section by Figs. 14, 15 and 16, consists of a cylindrical hollow piston gliding continuously on stainless steel strips on

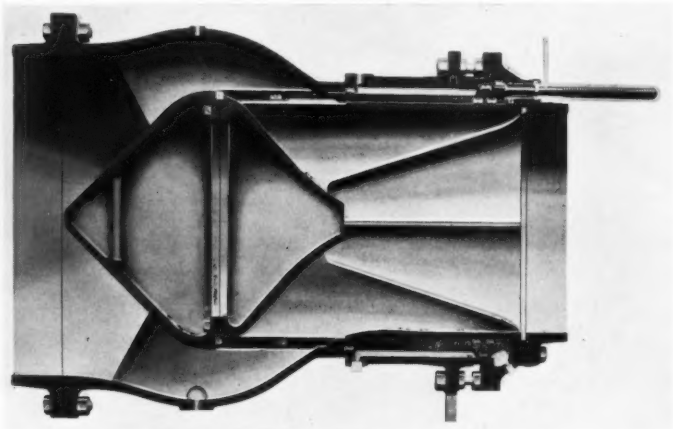


Fig. 14.—Diagrammatic section of Cylindrical Balanced Valve in open position.

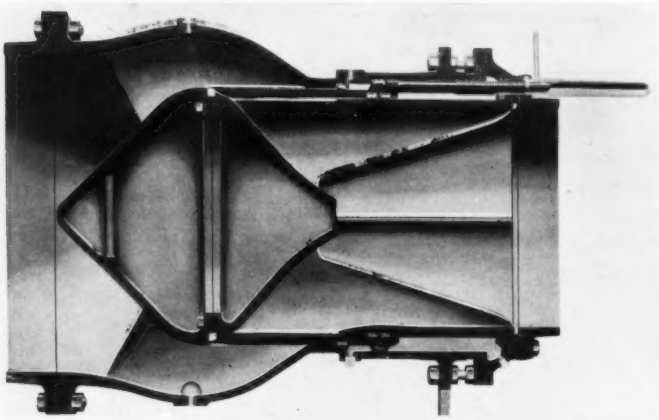


Fig. 15.—Diagrammatic section of Cylindrical Balanced Valve in closed position.

an inner body provided with ribs.* The valve piston ends in a head which moves axially inside the servo-motor, operated with filtered water under pressure. The radial and axial pressures on the piston are evenly distributed and self-balancing. The valves operate either automatically or by hand and remote control, by admitting the water pressure on either side of the servo-motor. The valve piston closes circularly against a neck ring embodied in the inner stationary body, which forms a streamlined smooth annular water passage with the outer body when the valve is in its open position. The balanced principle in design renders the valve inherently free from slam shutting and opening, thus obviating dangerous water hammer. The design so far described would result in opening and closing at constant speed. In order to reduce the operating times and make the latter adjustable, without increasing the pressure rise in the pipe line, a desirable requirement for all cases of emergency, a simple device in the form of a control diaphragm, Fig. 17, is fitted rigidly to the valve piston and travels with it. This travelling diaphragm is in the form of a threaded rod with variable pitch and depth of thread, working within a hollow cylindrical chamber so that the pressure water encounters on its passage into the servo-motor a flowing resistance which increases with the degree of closure. It is thus possible, in a simple manner, to obtain an asymptotic closing and opening movement, as shown by curves in Fig. 18, so that the valve can be made to operate rapidly over its full stroke and nevertheless close and open very gently, thus obviating pressure rise. This also completely obviates the necessity of a bypass for priming.

Experience has shown that the valve is drop-tight in service and that due to the streamlined design the friction losses are negligibly small. The valve embodies also a dismantling ring, thus rendering a special telescoping joint for dismantling unnecessary. This circular valve can

* *See "The Engineer," 1927, vol. 144, p. 418; also "Engineering," 1928, vol. 125, p. 771; vol. 126, p. 190; and 1934, vol. 138, p. 640.*

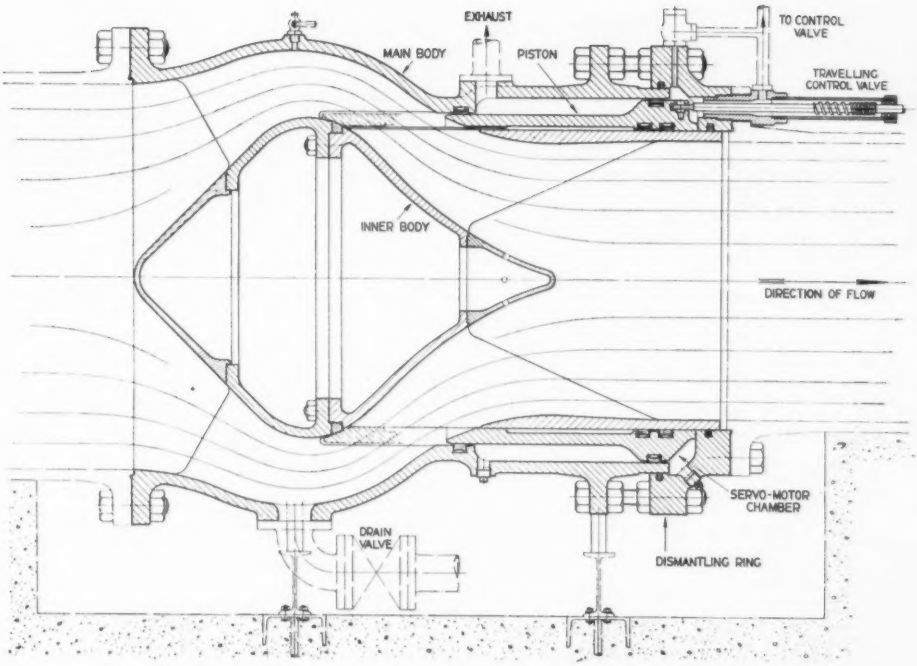


Fig. 16.—Section of Cylindrical Balanced Valve showing streamlining.

conveniently be arranged underneath the power station floor, so saving valuable floor space round the machine. It is free from wear and vibration.

Used in the opposite direction of flow, the valve becomes a relief valve or discharge regulator, Figs. 19 and 20.* It lends itself very conveniently to the purpose of guiding the water in a straight path from the turbine casing to the tailrace. By dispersing the water in a hollow cone, which quickly loses its energy, no harm is done to the tailrace, its foundations or other surroundings. The design of the valve lends itself conveniently to automatic operation in connection with the governor and turbine gears. It forms a correct cylindrical nozzle for any degree of opening and thus produces a smooth discharge for both small and large strokes. The synchronism with the turbine and governor, both quantitatively and in time, is very simple and permits of the plant being operated with a minimum pressure rise in the penstock.

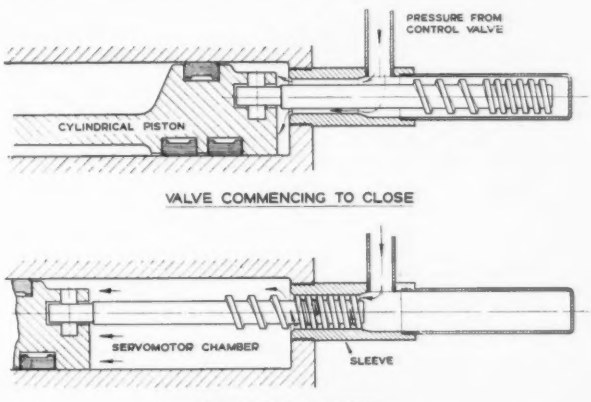


Fig. 17.—Travelling Control Diaphragm.

* *See "The Engineer," 1931, vol. 152, p. 218; also "Engineering," 1934, vol. 138, p. 640.*

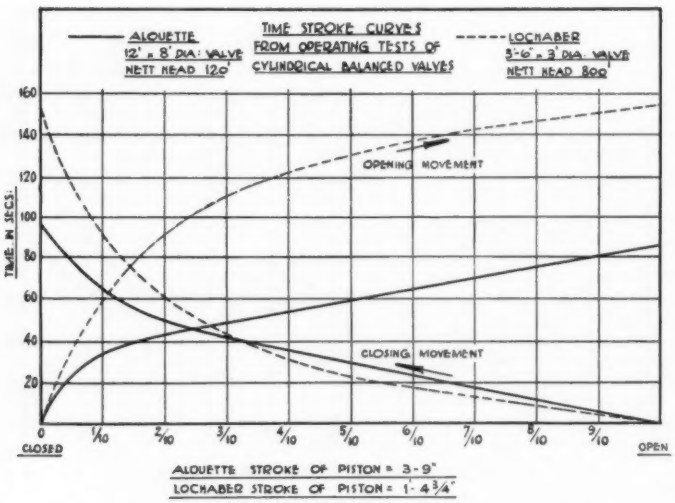


Fig. 18.—Asymptotic opening and closing time characteristics.

Apart from the use of the valve as a relief or bypass on the turbine proper, it is also in use as a discharge regulator on the dams of reservoirs, where it can conveniently be arranged for hand operation or for remote control.* Figs. 21 and 22 show a characteristic example of its use in this case.

The Hydraulic Brake.

The desirability of a reliable brake, with which the generating set can be brought to rest rapidly and safely without producing any heat or danger to the machine or operating staff, is fully appreciated. It is by no means necessary solely in a case of emergency, for it is well known that journal and thrust bearings especially suffer most if subjected to slow running down of the rotating element. Now the deceleration of turbine sets, embodying, as in most cases, very considerable moving masses, is very slow, so that large sets often take from 15 minutes to 1 hour to come to rest. Mechanical brakes can be made, but they are cumbersome and expensive; they always develop considerable heat and sometimes even cause sparks with dangerous consequences. Mechanical brakes are generally not allowed to be thrown into action at any speed because of the danger of causing excessive wear and heat. In case of an actual runaway the efficacy of the usual mechanical brake is therefore very problematic. If the mechanical brake is in daily, or almost daily, use for the routine stopping of the set as a safeguard for the bearings, the brake lining and drums require relatively frequent renewal, causing interruption of service and increasing maintenance costs. On machines with horizontal shafts the adoption of mechanical brakes is especially undesirable as they lengthen the distance either

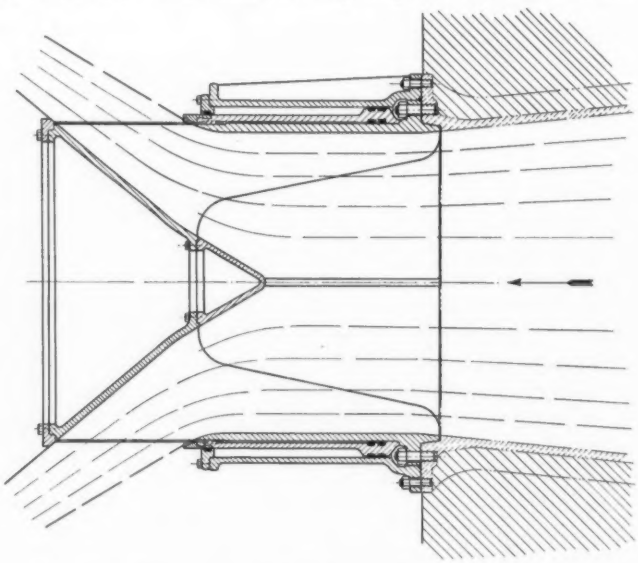


Fig. 19.—Diagrammatic section of streamlined Cylindrical Balanced Relief Valve.

* *See "Engineering," 1933, vol. 136, p. 681.*

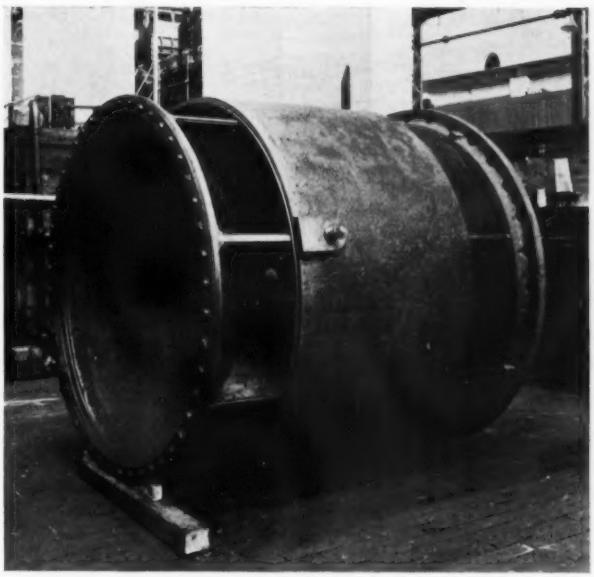


Fig. 20.—Shop assembly of Cylindrical Balanced Relief Valve.

between bearings or between turbine and bearings, thus adding to the cost by necessitating increased shaft diameters.

These considerations led to the development of the hydraulic brake, by means of which reaction turbine sets up to the largest sizes, both of the horizontal and vertical shaft types, can be safely brought to rest in record time regardless of the size of the moving masses or the speed, and without production of heat or wear. The principle of hydraulic braking by means of jets impinging on the runner wheels of impulse turbines in the opposite direction to the rotation is well known. The application of hydraulic braking was somewhat more difficult in the case of reaction type turbines.

A similar device to the single braking jet impinging on the back of impulse wheel buckets was, of course, of no use, and it was necessary to create a powerful circular braking vortex impinging on the back of the runner wheel blades at a certain angle without interfering with the normal operation or the balance of the runner or its strength. The arrangement is illustrated in Figs. 25 and 26. Reaction turbines of any size can be brought to rest within two or three minutes without any inconvenience by simply utilizing some pressure water from the penstock. The operation of the hydraulic brake can be made automatic by hand or by remote control. It requires no additional space and simply utilizes the runner wheels as brake drums. Fig. 28 shows deceleration curves for an 8,000 h.p. turbine with and without hydraulic brake in action.

The Impulse Wheel Jet Disperser Governing Device.

The usual systems of governing turbines of the impulse or Pelton type are by nozzle and bypass valve, or by nozzle and deflector. In the first system, the nozzle, which forms the working jet,

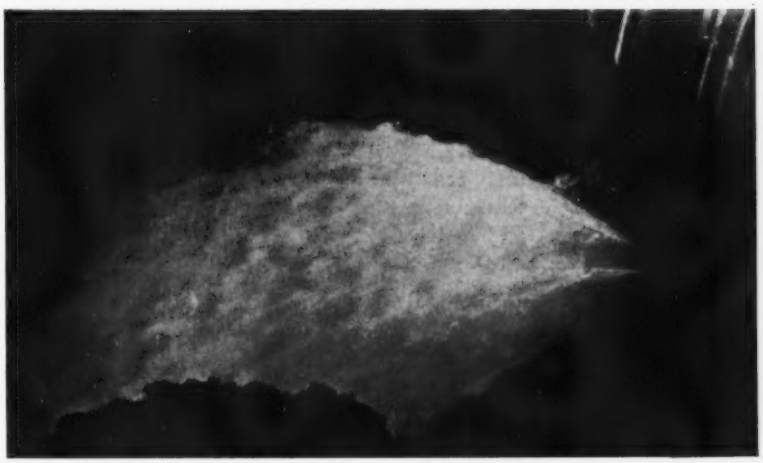


Fig. 21.—Spreading discharge of a Diffusing Discharge Regulator.

Fig. 22.—Discharge Regulator in operation on Mahinerangi Dam, N.Z.

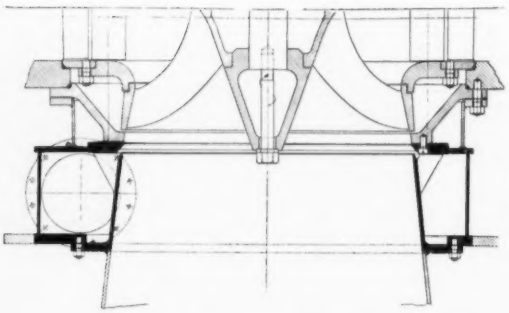


Fig. 23.—Vertical shaft arrangement of Hydro-Brake.

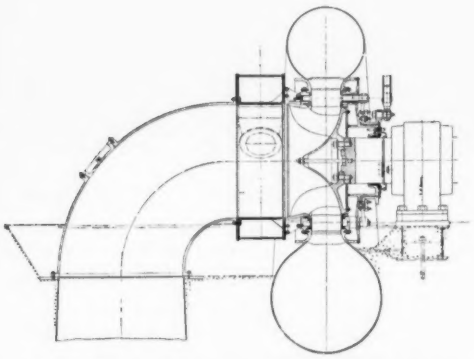


Fig. 24.—Horizontal shaft arrangement of Hydro-Brake.

is rapidly closed and a bypass or discharge regulator simultaneously opened so as to avoid excessive water hammer in the pipes by compensating the water cut off from the turbine. To obtain reasonably close governing, considerable operating forces must be set in action, requiring in turn cumbersome mechanisms. The size of the governor must be adapted in each case according to the head and water quantity prevailing in the plant. The rapid handling of large jets is always difficult and risky since the momentum of quantities of moving water is very great. In

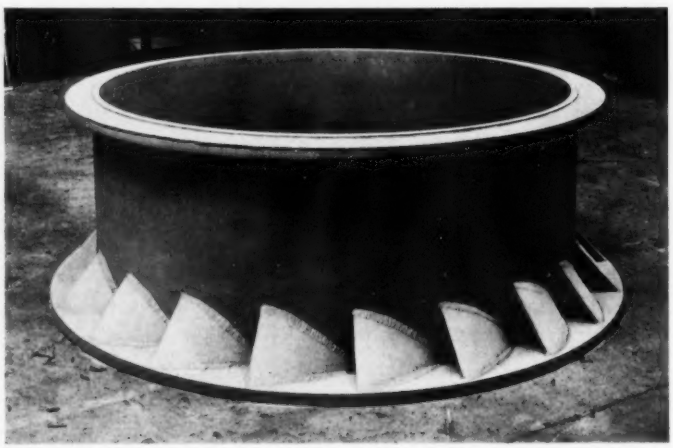


Fig. 25.—Nozzle piece for Hydro-Brake for Reaction Turbine.

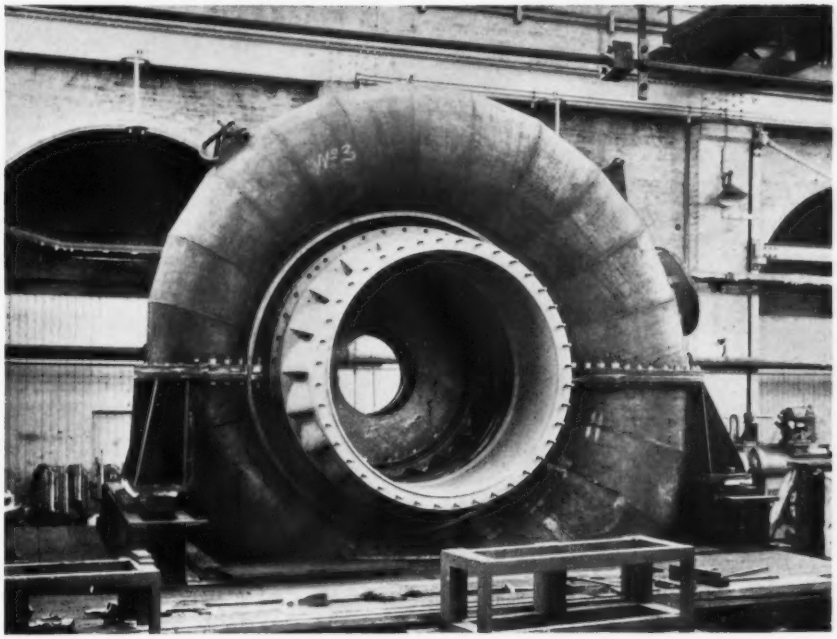


Fig. 26.—Brake Nozzle Piece assembled on Reaction Turbine.

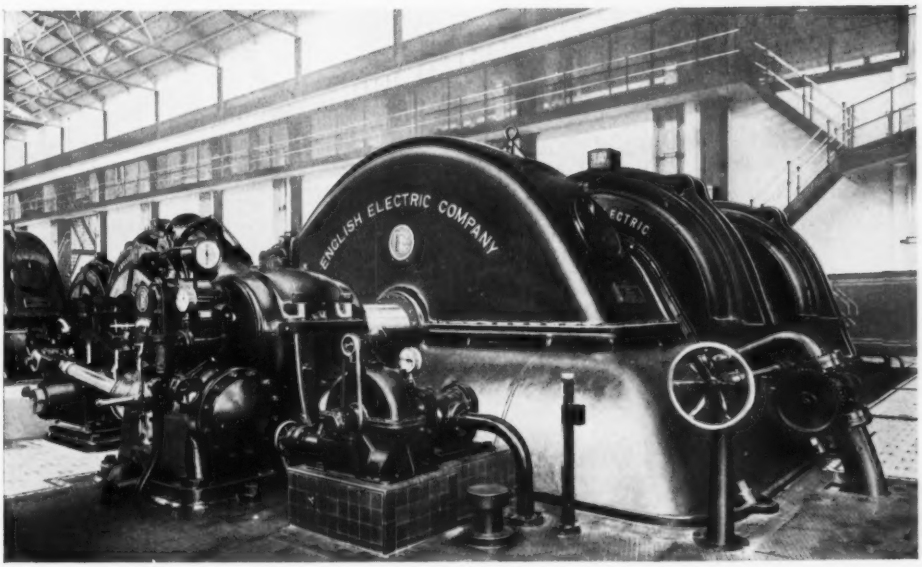


Fig. 27.—Hydro-Brake (on right-hand side of illustration) adapted to Impulse Turbine.

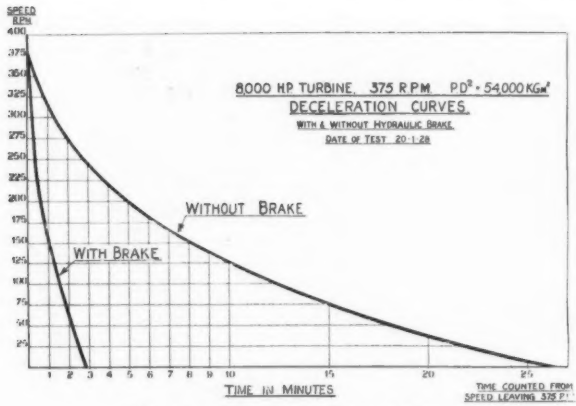

SPEED R.P.M.
8000 HP TURBINE. 375 R.P.M. PD² = 54000 KGm²
DECELERATION CURVES.
WITH & WITHOUT HYDRAULIC BRAKE.
DATE OF TEST 20-1-28
WITHOUT BRAKE
WITH BRAKE
TIME IN MINUTES
TIME COUNTED FROM SPEED LEAVING 375 R.P.M.

Fig. 28.—Diagram of Brake Action.

medium-and low-pressure plants, where combined bypass governing is resorted to, the pipes can generally be designed to withstand an abnormally increased pressure rise in case of failure of the relief or bypass valve. In high-pressure plants, however, the provision of pipes of greater thickness to increase the safety factor is generally fraught with considerable difficulties, both technical and economic. For this reason the bypass system of governing is inadvisable for high-pressure plants, quite apart from the fact that the results, in spite of the large and expensive mechanisms required, are by no means perfect.

Combined Double-acting Nozzle and Deflector Governing Device.

This circumvents the necessity of practically slam-shutting the nozzle in case of a drop in load, but for close governing it also requires very considerable forces to deal rapidly with the large quantities and high heads. The size of the governor must in this case also be adapted to the particular plant and, in addition, an automatic and quite intricate relationship must exist between the size of the jet for each load and the position of the deflector. This is vital in order to avoid "hunting," because the primary or deflector system must be made to act on any given size of jet without lost motion. It is the English Electric Company's practice to provide combined nozzle and deflector governing systems on plants which do not require particularly close governing, such as the Lochaber turbine shown in section in Fig. 29.

In most high-pressure plants, however, the governing system must provide very close governing in case of a drop in load and yet be simple in operation, whilst ensuring absence of wear and maximum safety under all conditions of service. Efforts were therefore made to devise a governing system requiring very small forces for its operation, thus permitting of a universal governor for all impulse wheels up to the largest sizes. The well-known conical nozzle and needle are carefully designed for the formation of a compact cylindrical jet so as to ensure maximum efficiency. The size of the jet is regulated in the usual way by means of a needle or spear moving axially inside the nozzle, see Figs. 31 and 32. It was observed that even slightly conical or "broomy" jets rapidly diminished the efficiency of the jet and turbine. This phenomenon was used in the invention of the diffuser governing system.* This

* "*Engineering*," *1927, vol. 123, p. 317.*

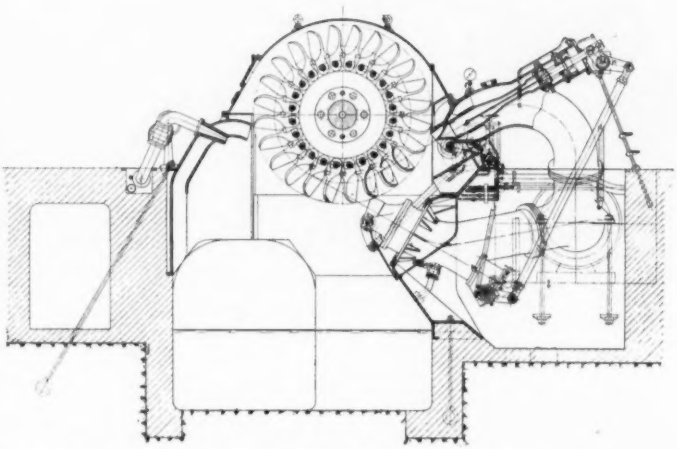


Fig. 29.—10,000 h.p. Double-Nozzle Impulse Turbine for Lochaber.

Fig. 30.—The Lochaber Hydro-Electric Power Station of the North British Aluminium Company, containing five 10,000 H.P. Water Turbines driving D.C. generators having an aggregate output of 100,000 amperes.

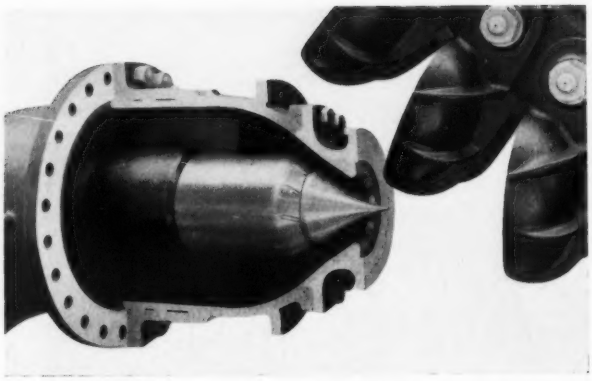


Fig. 31.—Needle Shaft and Diffusing Blades in normal position under load.

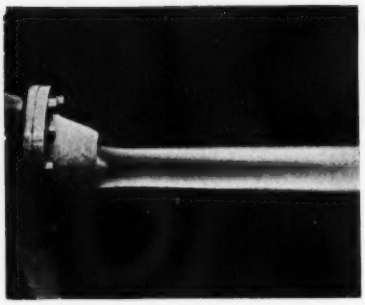


Fig. 33.—Jet operating under normal conditions.

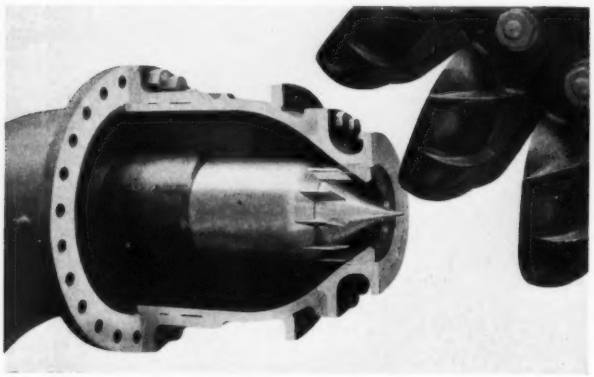


Fig. 32.—Needle Shaft with Diffuser in acting position.

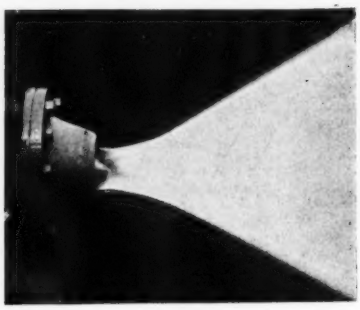


Fig. 34.—Jet with Diffuser in action.

consists in dispersing or diffusing the jet by means of a number of slightly slanting and streamlined guide vanes surrounding the needle, which, when protruding from the needle, as when governing, impart a slight rotary motion to the jet. This initial vortex is enormously enhanced by the transformation of the potential energy into kinetic energy which takes place inside the nozzle, with the result that the jet outside the nozzle is rapidly and completely broken up into a hollow cone, whereby the energy imparted to the runner wheel is very rapidly destroyed. In normal operation the diffuser plates are permanently withdrawn inside the needle, thus permitting the formation of an efficient cylindrical jet. The initial vortex can be brought about by comparatively small forces even for turbines of the largest sizes and for very high pressures, and diffusion takes place almost instantaneously, resulting in very close governing. Figs. 33 and 34 show respectively a set operating under normal conditions and with the diffuser in action.

This system of regulation has now been in continuous commercial operation for many years and is giving consistently good results. No detrimental influence on the pipe line pressure occurs during its operation, and no wear whatever takes place on the diffuser plates. In fact, there has never been occasion to renew or replace any diffusers.

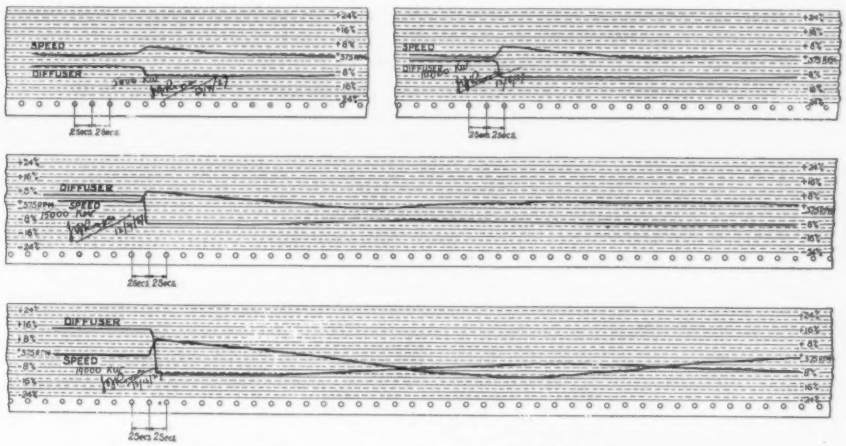


Fig. 35.—Tachogram Curves of Governor with Jet Disperser.

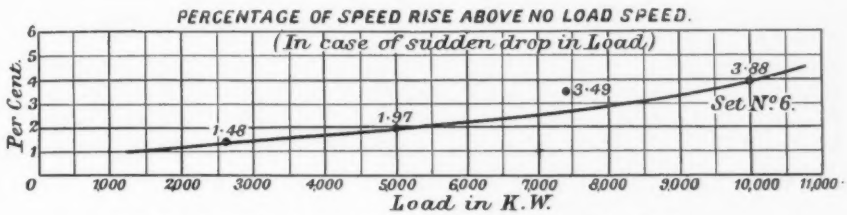


Fig. 36.—Operating characteristic of Jet Disperser Governing Device.

Fig. 35 shows some results obtained on an important installation, and Fig. 36 shows the operating characteristics of the governor. No special protection in turbine pits or tailraces is necessary because the energy of the diffused jet is largely destroyed when it reaches these points. Finally, no system of interdependence between the size of the jet and the position of the diffuser is required, because the diffuser, after action, always withdraws to the same position inside the needle flush with the latter, and is thus ready to act instantly on any size of jet without lost motion.

Impulse Wheel Bucket Fastening.

The dynamic forces and stresses prevailing in impulse wheel buckets and their attachments present a problem which, apart from conditions encountered in ordnance work is, the author believes, unique in engineering practice. A satisfactory bucket fastening must afford absolute security under all conditions of service and ensure perfect interchangeability and easy and rapid handling in the field.

For the attainment of high efficiency and freedom from wear it is necessary to design the form of the buckets by determining the relative paths of at least a number of the most important drops inside the rotating buckets. This is done by the judicious application of Coriolis's theorem of relativity, and is corroborated by frequent tests in the field. The size and weight of buckets for large machines are often considerable.

The attachment of the buckets to the wheels or discs must be designed and manufactured to withstand :—

(1) The alternating stresses due to the jet impact.

(2) The centrifugal forces at normal and runaway speeds.

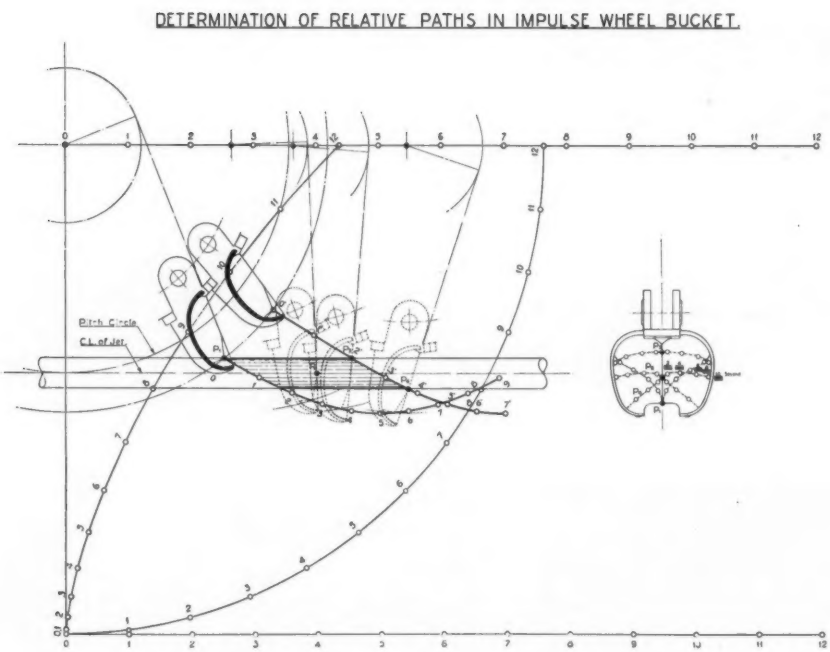


Fig. 37.—Determination of Relative Paths in Impulse Wheel Bucket.

(3) The stresses set up by the application of the full jet with the wheel at rest (short circuit).

An example will best illustrate the method of design. For a wheel developing 15,000 h.p., for instance, and running at 375 r.p.m. under 1,650 feet head, each bucket sustains a thrust from the jet amounting to 27.5 tons 6.25 times per second, or 540,000 times per day. Fig. 37 shows how relative paths are determined, and it will be seen that each particle of water traverses the bucket in about $\frac{1}{100}$ second. The thrust of the jet, therefore, is decidedly in the nature of a blow. Each bucket, after passing through the jet, is completely emptied again so that the resultant forces alternate continuously and very rapidly. In addition, there are the centrifugal forces set up concurrently by the rotation. It will thus be realised that the demand on the attachment is very severe and it is not surprising that numerous bucket attachments by several cylindrical and also conical bolts, rivets, or even dovetailed edges, have proved inadequate, resulting in dislocation and rupture with disastrous consequences.

The system of attachment adopted by the English Electric Company* consists of one bolt only for each bucket, made up of a bush in nickel-chromium steel, cylindrical outside and conical inside, which is split over its whole length. A conical bolt, also of nickel-chromium steel, is introduced into the bush, and is provided on either side with a fine thread, nuts, and locking washers. The holes in the bucket and the disc are cylindrical and of identical diameter. This facilitates manufacture and enables the attachments to be completely finished in the shops so that the buckets become strictly interchangeable, chiefly because each has only one hole. On introducing the conical bolt into the split bush after assembling the bucket with the disc, the split bush expands, and is caused by the cone to bear against the cylindrical holes in the disc and bucket with a tightening force considerably superior to any

* *See "Engineering," 1928, vol. 125, p. 747.*

force the jet can exert on the bucket. No dislocation or weakening of the attachment can therefore take place, and in practice no bucket attachment of this type has ever shown any dislocation or similar trouble. Every alternate bucket is made to bear against a part protruding from the disc and cast integral with the latter by two strong transverse (axial) keys, each of which is separately locked. The conical side or taper is arranged between the two keys so that each bucket presents parallel facings only, thus again ensuring perfect interchangeability. The method of attachment is illustrated in Figs. 38 and 39.

Dismantling is easy and expeditious as the keys can be separately withdrawn as for an ordinary wheel and shaft attachment, while the bucket can be slackened by simply withdrawing the conical bolt from the split bush by means of its threaded end and nut. No reamering is necessary on site and thus prolonged stoppage of the plant is obviated in case of replacements.

Experimental and Research Plants.

In every hydro-electric scheme nature imposes peculiar conditions of pressure, output, quantity, and local setting. Hydro-electric engineering is thus to a large extent pioneer work, and it is often necessary to resort to extensive experimental and research work in order to achieve the best results. The English Electric Company has thus been led to establish its own industrial test plant, which is, as far as the author knows, the first and only one in the British Isles, apart from the experimental laboratories in technical schools, the purpose of which is chiefly educational.

This plant, which is illustrated by the photographs and sectional drawings reproduced in Figs. 44 to 49 inclusive, enables extensive tests on model turbines and their accessories to be made. It is established within the factory area so that research can be carried out in constant contact with the design and manufacturing staffs.

By means of a test plant the efficiency of turbine runner wheels covering a wide range of specific speeds can be established without ambiguity. Such a plant enables the correct form of spiral casings, distributing apparatus, and draught tubes to be determined. It can also furnish valuable information on axial thrusts, cavitation, runaway speeds, and other hydrodynamic phenomena

Fig. 38.—Method of attachment of Impulse Wheel Buckets.

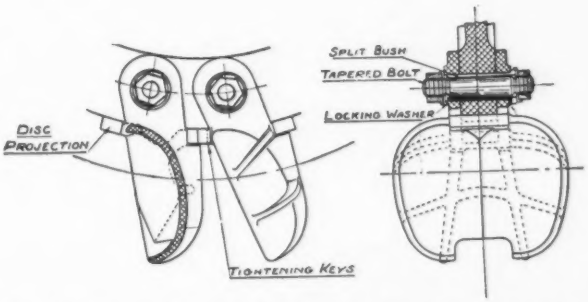


Fig. 39.—Sectional view showing method of attachment of Impulse Wheel Buckets.

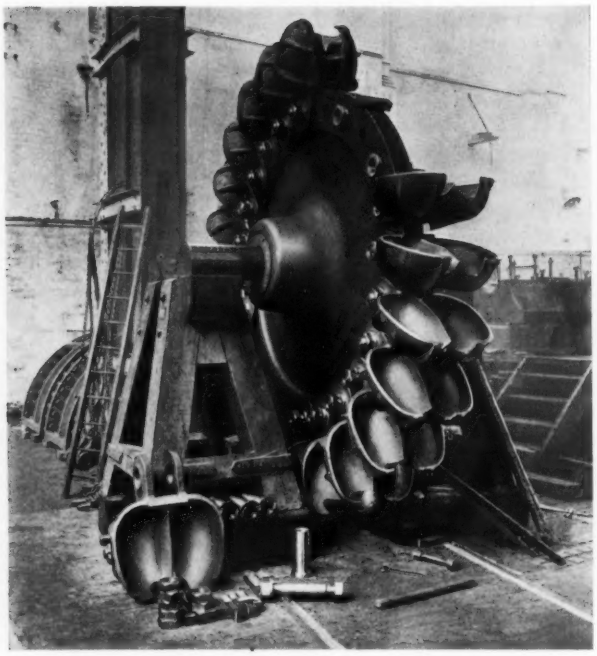


Fig. 40.—15,000 h.p. Impulse Runner in course of erection.

which it is very difficult, if not impossible, to establish theoretically with sufficient accuracy. It is possible to establish the deceleration characteristic resulting from the application of hydraulic brakes as compared with the unbraked deceleration, and to determine the quantity of water required for a given braking effect. The characteristic of the aerofoil type of flow recorder can likewise be exactly determined.

The work also includes a thorough testing of all governors and their safety devices, valves, discharge regulators, oil pressure plants, and measuring instruments; the testing of pressure-bearing parts, and the concurrent determination of the properties of the materials employed; runaway tests, and tests of the static and dynamic balance of runner wheels, flywheels, and other important rotating elements.

The original plan to utilize the water level difference of an existing river was dropped in order to enable the tests to be made independently of fluctuations in quantity and head, often considerable in any river. Consistent and reliable tests under steady conditions can only be obtained by means of pumped water and tanks which allow of exact adjustment of quantity and pressure. This also made it possible to install the experimental plant within the works. The water circulation plant consists of a vertical-shaft axial-flow pump directly coupled to a vertical-shaft 130 h.p. electric motor permitting variable speed. The pump can deliver a maximum of about 1.2 tons of water per second against a head of 20 feet. The

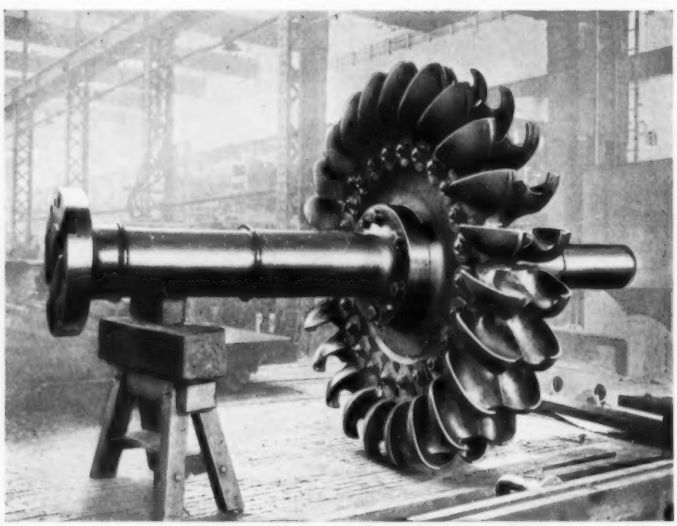


Fig. 41.—9,600—10,000 h.p. Impulse Turbine Runner and Shaft for Lochaber, Scotland.

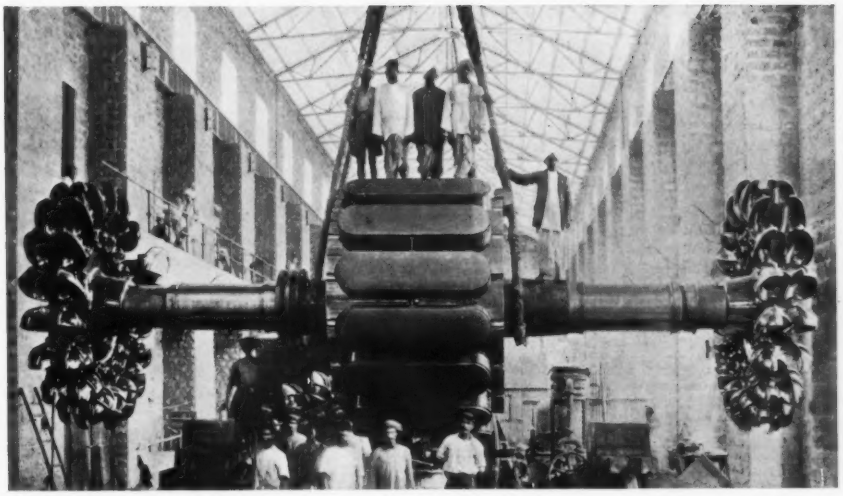


Fig. 42.—30,000 h.p. Impulse Turbine, Twin Runner and Alternator Rotor for Bhira, India.

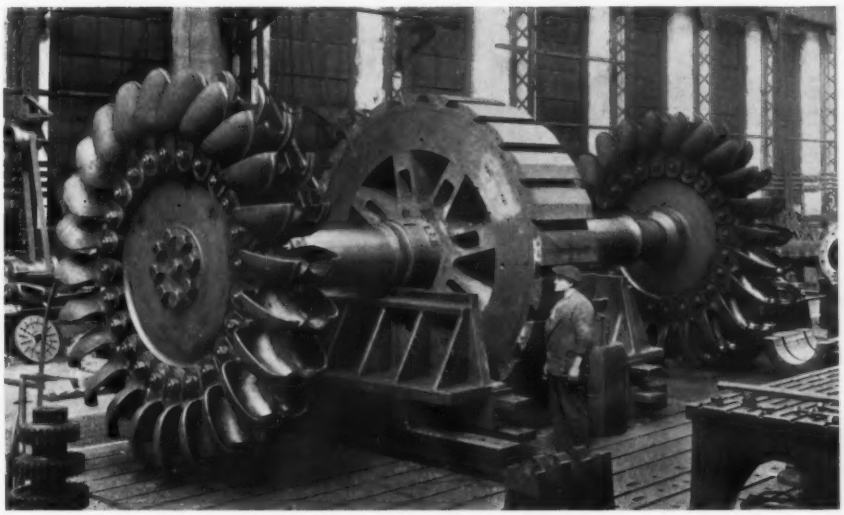


Fig. 43.—18,000 h.p. Impulse Turbine, Twin Runner and Alternator Rotor for Jordan River, B.C.

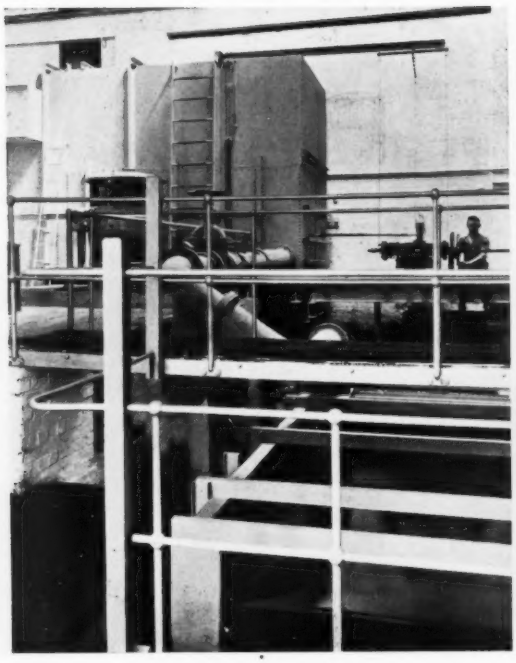


Fig. 44.—Elevation of Hydro-Test Station, Rugby. Calibrated Weir.

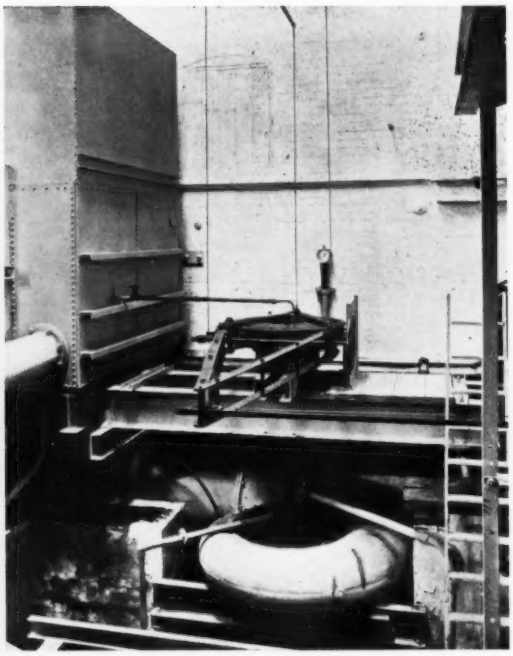


Fig. 45.—Elevation of Hydro-Test Station, Rugby. Spiral cased Reaction Turbine and Test Brake.

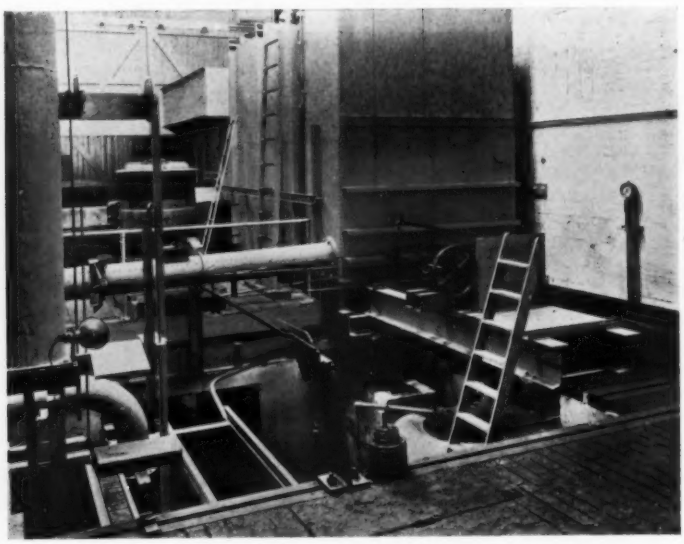


Fig. 46.—Test Station with Gauging Levels and Instruments.

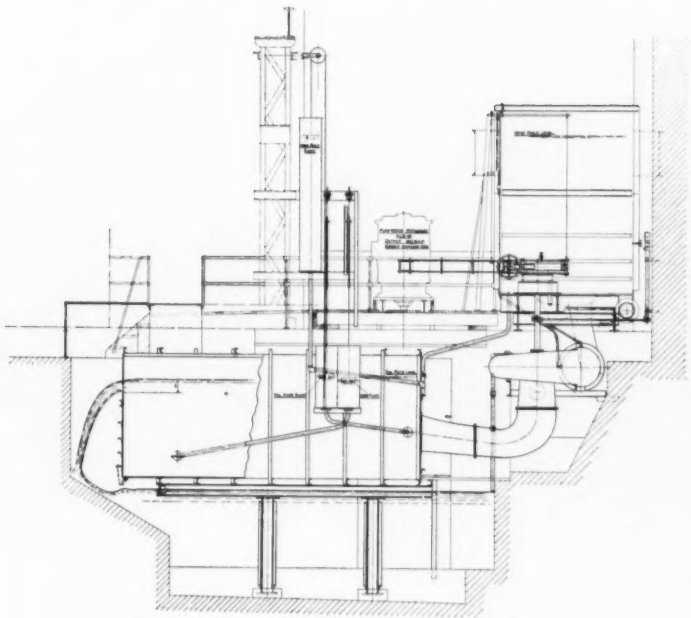


Fig. 47.—Elevation of Hydro-Test Station, Rugby.

quantity and head utilized by any turbine under test can be adjusted by means of the speed of the pump, in conjunction with a head tank provided with an overflow weir, into which the water is delivered by the pump. A very steady level of water in the head tank is secured by means of a number of streamlined flat wooden plates reaching from top to bottom of the tank at intervals of about 1 inch. The water is conducted to the spiral casing and turbine to be tested by means of a pipe and bend and reaches the tailrace through a suction pipe and bend, easily accessible and interchangeable, to enable various forms of suction bends to be tested. All forms of spiral casings can be tested and the layout can be adapted for testing various forms of guide and distributing apparatus, runner wheels of every form and description, hydraulic brakes, shutters, etc.

The turbine shown in Figs. 44 to 49 is of the vertical-shaft type, but horizontal-shaft turbines can also be adapted and tested. The tailrace is arranged as a gauging tank and is provided with a calibrated rectangular contraction weir by which the water returns to the sump at the bottom of the building where the circulation pump operates. The vertical-turbine shaft rotates in two ball bearings and is suspended in a thrust bearing of the ball race type.

The power developed by the test turbine is absorbed by a mechanical and compensated friction brake of the Prony type provided with three equidistant pivoting brake shoes with Ferodo linings bearing against a fabricated circular drum constantly cooled by water from the inside. The torque is measured by adjustable counterweights attached to a cable connected with the arm of the brake. It is thus possible to determine the power delivered by the turbine

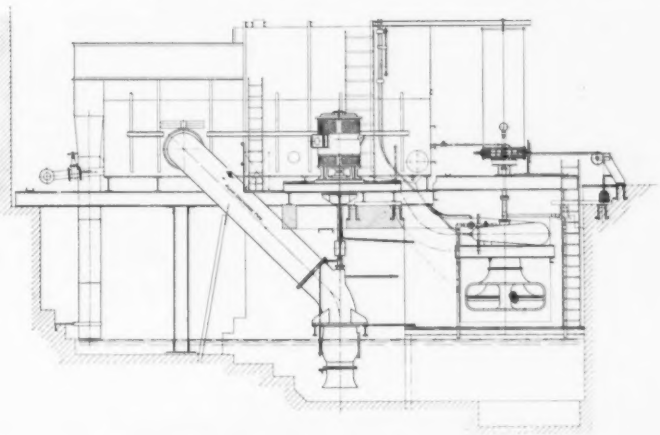


Fig. 48.—Elevation of Hydro-Test Station, Rugby.

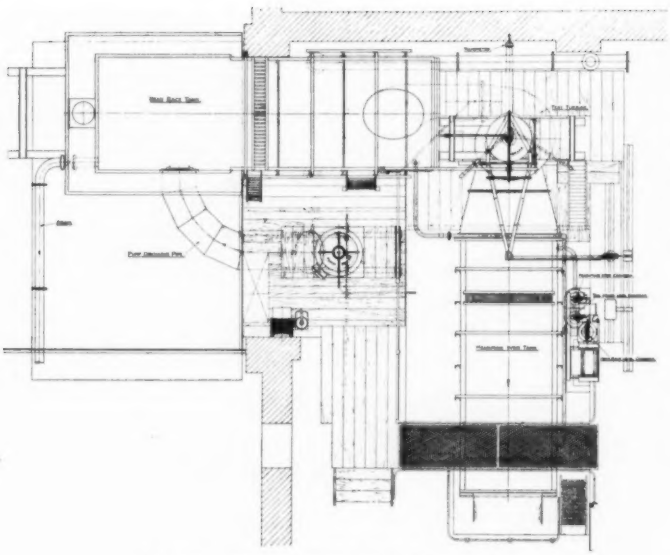


Fig. 49.—Plan of Hydro-Test Station, Rugby.

under any conditions of load, head, and speed, with considerable precision. The effective head is measured by float indicators in head and tail tanks and the various losses in bends, spirals, and all other vital parts are measured by piezometers. The speed is measured both by a permanent and a belt-driven dial tachometer and is controlled during tests by a hand tachograph.

Sample test results obtained with this plant are shown in Figs. 50 and 51. The tests made, especially the efficiency tests, have always shown excellent agreement with the results obtained from the full-size machines in the field. It is, however, necessary to apply correcting factors to take into account scale effect. The money invested in the test plants has been amply justified by the invaluable aid it has afforded in devising improved forms and elements of turbines and their accessories.

General Design and Special Mechanical Features in British Practice.

One of the most pronounced features of British practice is the comprehensive way in which hydro-electric schemes are being developed. While the turbines, of course, as prime movers, primarily determine the type, speed, local setting, connection, and operation of the generators, constant contact between the hydraulic and electrical engineers allows the machinery to be treated as a whole "hydro-electric set." From the start careful consideration is given to the adoption of output, type, and speed suitable to both the

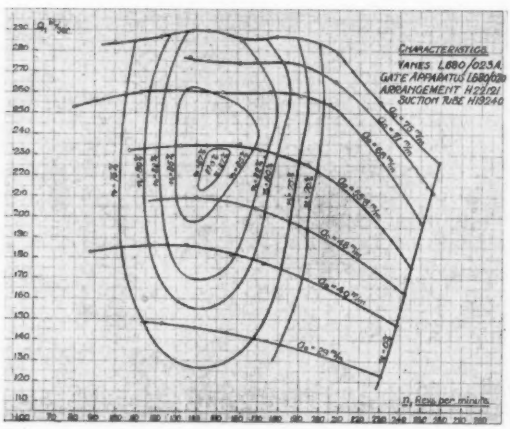


Fig. 50.—Test Result on Reaction Turbine.

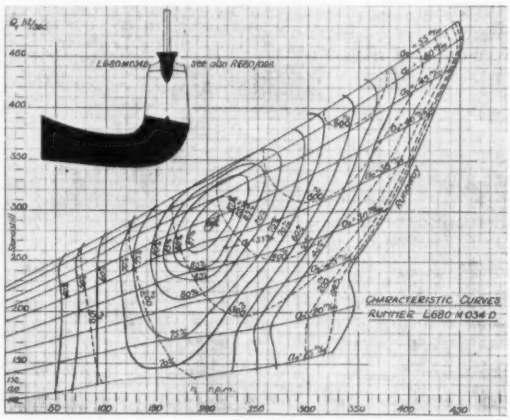


Fig. 51.—Test Result on Propeller Turbine.

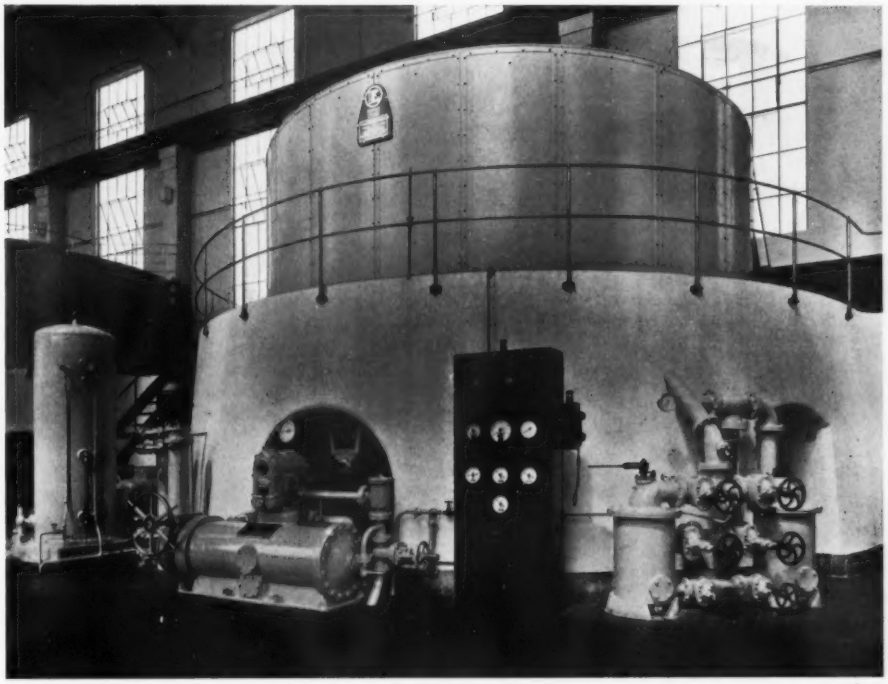


Fig. 52.—One of the 15,500 h.p. Generating Units in the Tongland Power Station of the Galloway Water Power Company.

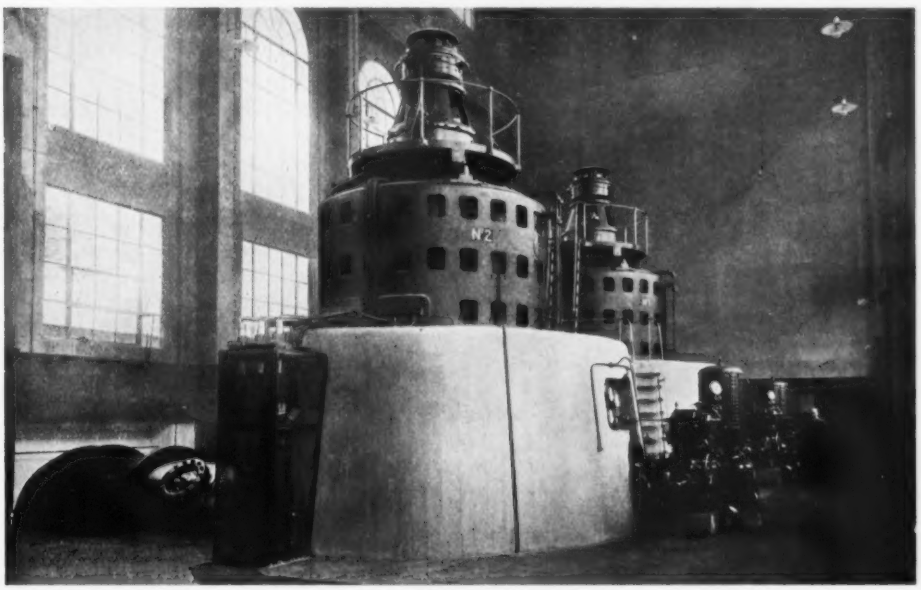


Fig. 53.—Two 4,000 h.p. Generating Units in Stonebyres Power Station of the Lanarkshire Hydro-Electric Power Company.

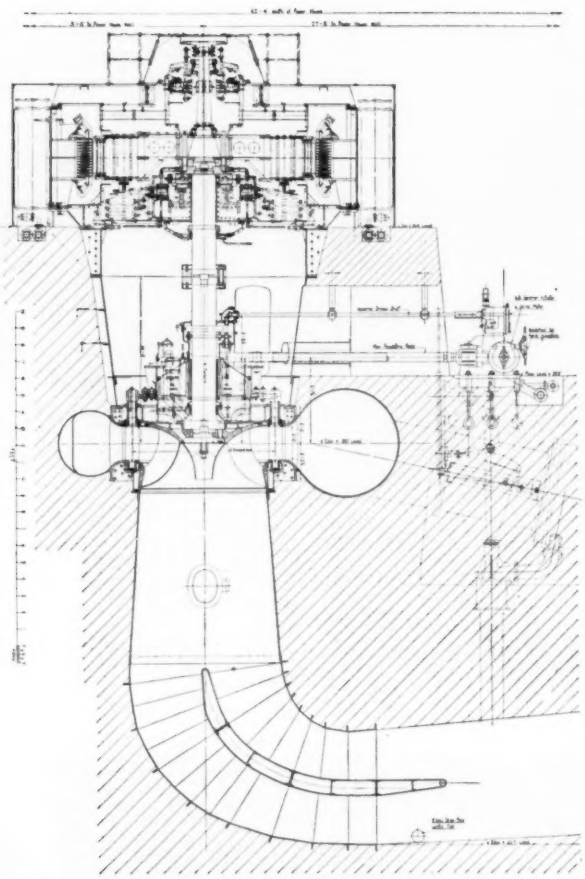


Fig. 54.—Sectional elevation of one of the 15,000 h.p. Generating Units for Kendoon Power Station.

hydraulic and electrical parts of the plant. Adequate factors of safety are aimed at for all parts of the running machinery under both normal and abnormal operating conditions, including overspeed and short-circuit. At this stage the flywheel effects which it is possible and safe to embody in the sets to ensure satisfactory regulating conditions are determined, taking into account the conditions imposed by the momentum of the moving water both in the pipe line and tunnel. Great care is taken to create permanently rigid settings for the machinery, and to ensure their perfect alignment, connection, and accessibility. It is then possible to prepare compact designs occupying minimum floor space, and the cost of the power house can be reduced, and control and safety gears can be made simple and reliable.

Comprehensive treatment also includes the other components of the plant. Thus control gears, feeder piping, stop valves, air ducts, discharge canals, cable trenches, auxiliary machinery like exciter and standby sets, interconnecting plant, crane, etc., can be evolved so as to combine economy with ease of supervision, handling, and maintenance. Sufficient dismantling space is carefully considered in relation to access roads, crane height, and capacity. Continuous collaboration with the electrical designers allows the switchboards, transformers, cable trenches, and other electrical gear to be arranged in such a way as to avoid interference with any other service.

The arrangement and size of penstock and distributing pipes and their influence on governing are carefully investigated. The power house foundations, anchor blocks, pipe supports, and expansion joints are designed in the light of all normal and abnormal conditions obtaining in service. Seismic loading, altitude, and climatic conditions and their influence on the aeration and cooling of generators and bearings are also taken into account. Chemical and physical analyses of the water form the basis for the choice of the material for the runners, distributing apparatus, and other vital parts. Such comprehensive treatment, both in design and manufacture, makes possible the adequate design of completely automatic and semi-automatic plants and their controls.

The Block and Step Design of Power House.

Except in certain cases where horizontal-shaft machines are preferred for reasons of local setting, existing foundations, or a special type of service, as, for example, the operation of large direct-current generators, the single-runner vertical-shaft

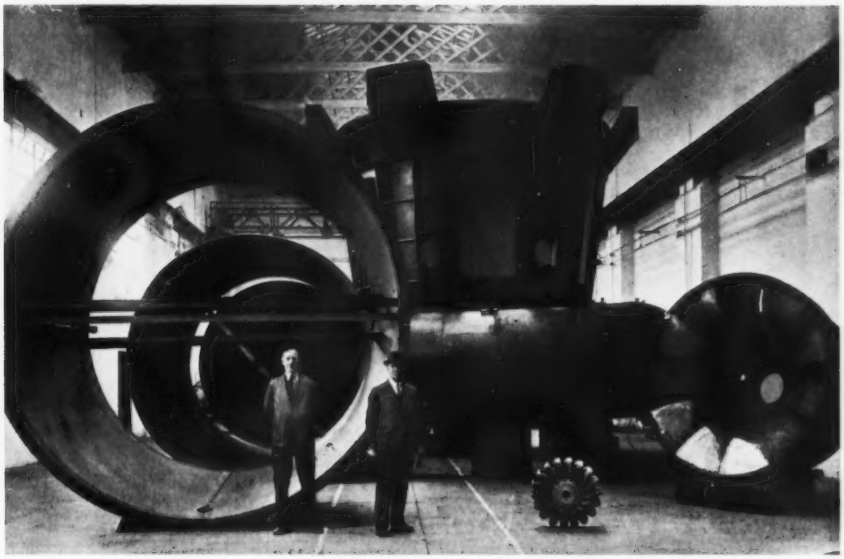


Fig. 55.—Shop assembly of 15,500 h.p. Turbine, Tongland.

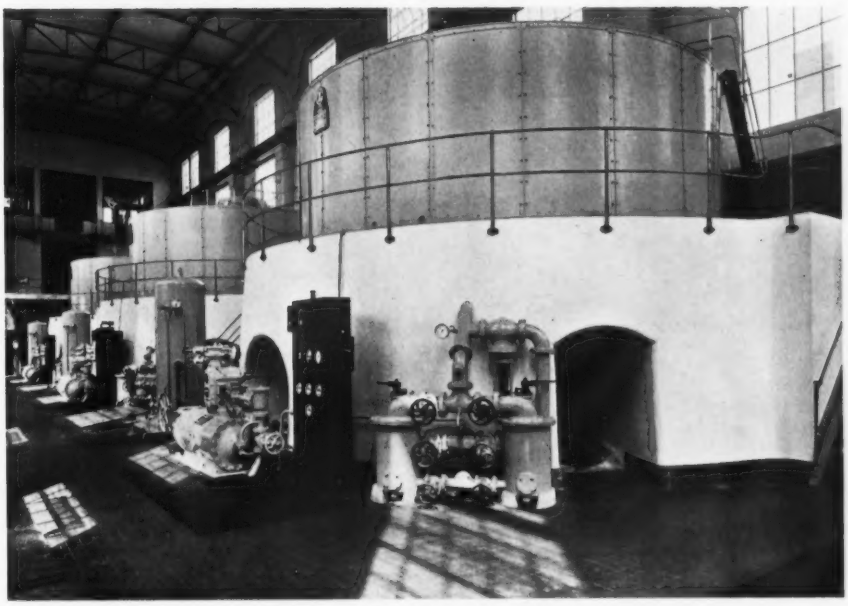


Fig. 56.—Interior of Tongland Power Station.

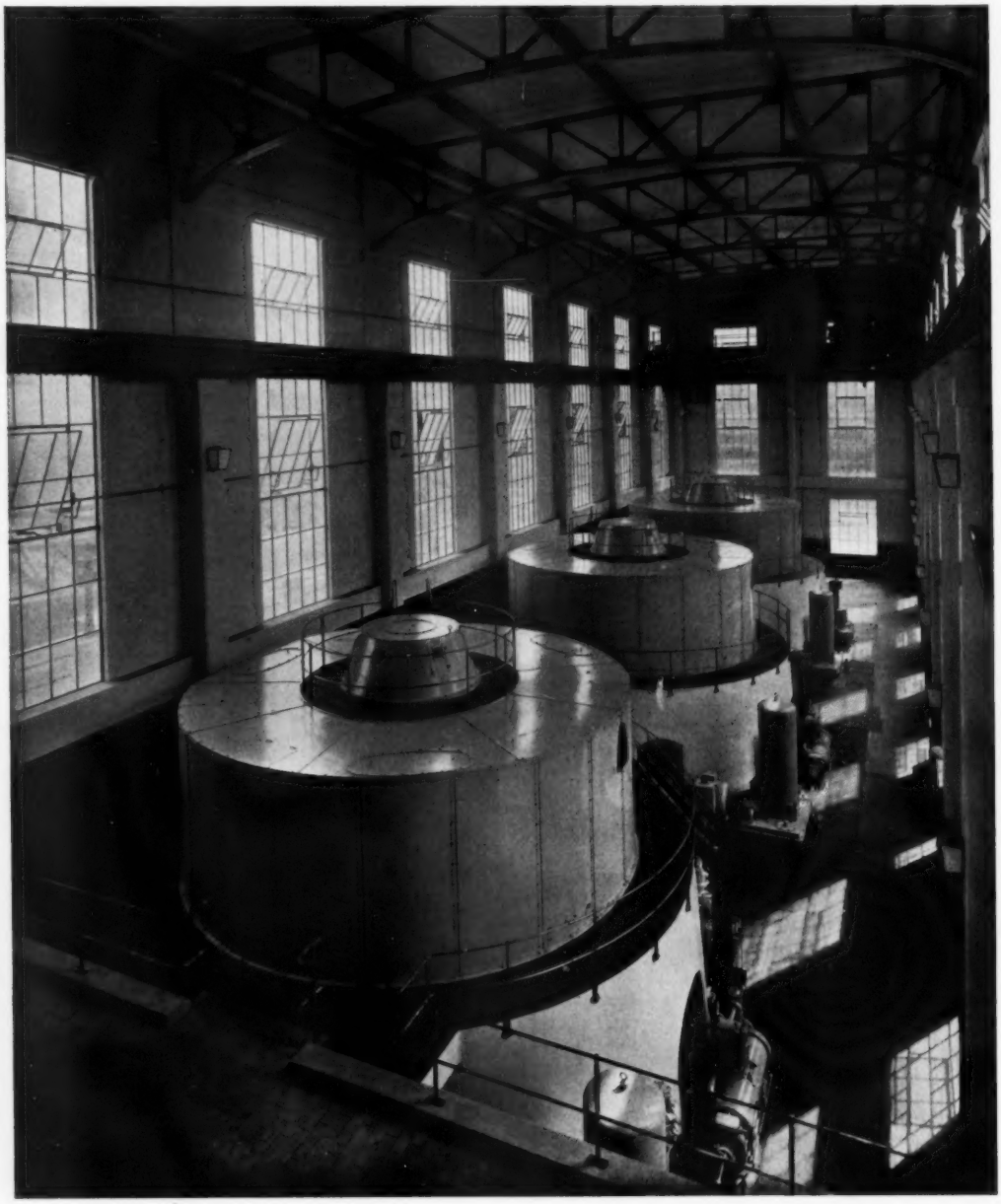


Fig. 57.—The Tongland Power Station of the Galloway Water Power Company, containing three 15,500 h.p. Water Turbine Driven Alternators.

type of reaction turbine and generator has been developed. This type of turbine ensures simplicity, sturdiness, economy of space and superior hydraulic conditions. The casing can be completely embedded in the concrete foundations, thus ensuring great rigidity and freedom from vibration. Hence the vertical-shaft type is given preference practically everywhere for machines of large output and size. The orthodox type of vertical-shaft plant involves, however, the following drawbacks. It is far more inaccessible than the horizontal-shaft plant. The usual design of vertical-shaft power house involves two or more floors, rendering dismantling, supervision, and maintenance awkward. Important parts like pumping plant, governors, and servo-motors must be arranged on different floors, often at a considerable distance from the parts they are destined to control, such as distributing apparatus, relief valves, and stop valves. Substantial interconnecting gears are therefore necessary in order to obviate lost motion and subsequent "hunting," the servo-motors must be embedded in the concrete, where they are inaccessible and difficult to maintain. A further objection is the difficulty of maintaining perfect alignment and rigid connections between the hydraulic and electrical machinery when they are arranged on separate and independent floors, because of the temperature differences and constantly varying axial loads.

To overcome these drawbacks a permanent and absolutely rigid connection between the turbine and generator was evolved, consisting of a steel barrel which is in reality nothing else than an extended generator bedplate. This barrel is permanently connected at the bottom to the turbine speed ring and is strongly embedded and anchored into a concrete pedestal surrounding the turbine. It rises from the power house floor, with which it is strongly interlaced, and is provided with three or more large access ducts traversing both pedestal and barrel. Through these ducts the turbine can be conveniently and rapidly visited for inspection from the single power house floor. In this way it is also possible to arrange the governor, servo-motor, and complete control mechanism in the immediate neighbourhood of the distributing apparatus and relief valve. The barrel transmits

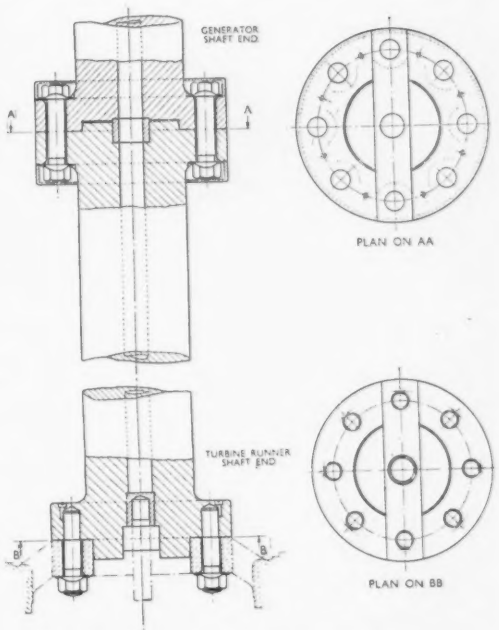


Fig. 58.—Turbine and Generator Shaft Couplings.

the weight of the generator through the cast steel speed ring of the turbine to the foundations underneath, and also takes up the short-circuit torque, which is a multiple of the normal torque. The ducts also allow the governor drive to be arranged in the most direct way. The cylindrical balanced stop and relief valves can conveniently be placed underneath the power house floor and their design greatly simplifies the foundations. All important parts are rendered easily accessible from the single power house floor and by the same crane. The oil pumping plant is also arranged on the same floor and in the immediate neighbourhood of the governor. The governor actuator can always be combined directly with the servo-motor underneath, that is, it can be of standard type. The inside of the turbine casing and runner is accessible through ducts and manholes from the same single power house floor.

The Coupling.

Perfect rigidity, concentricity, and alignment, combined with ease of manufacture, erection, dis-

Fig. 59.—Spiral Casings on foundations.

mantling, and interchangeability in the field, are desiderata in the design of a coupling to serve for the assembly of a turbine runner and shaft and of turbine and generator shafts. The usual multiple bolt type of coupling involving a number of cylindrical, or even conical, bolts was abandoned because of the considerable difficulty of manufacturing strictly interchangeable spare parts which would permit of expeditious replacement in the field without reamering. It is, indeed, practically impossible to repeat a series of, say, ten or twelve holes, which in respect of both diameter and pitch circle will correspond exactly, the more so as the necessary centring spigot must be made to fit simultaneously with a tolerance of not more than 1/1,000 inch or preferably less. Conical and cylindrical shaft and runner assemblies often render dismantling, after prolonged service, extremely difficult and can lead to prolonged stoppage in case of replacement.

In order to obviate these difficulties a coupling has been designed, as a standard product, made up of a number of studs or bolts separately and jointly locked, which are arranged within holes in the coupling and allow considerable clearance.

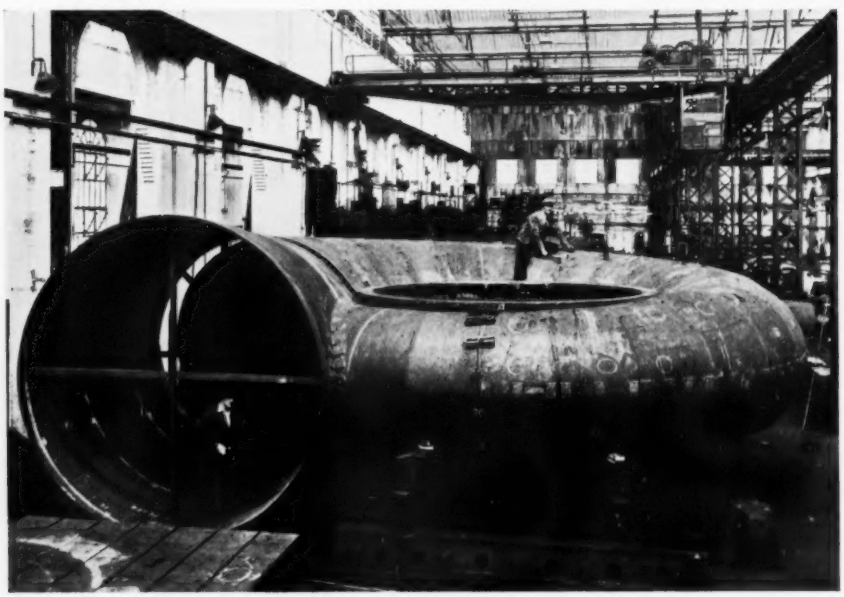


Fig 60.—Spiral Casing in shops.

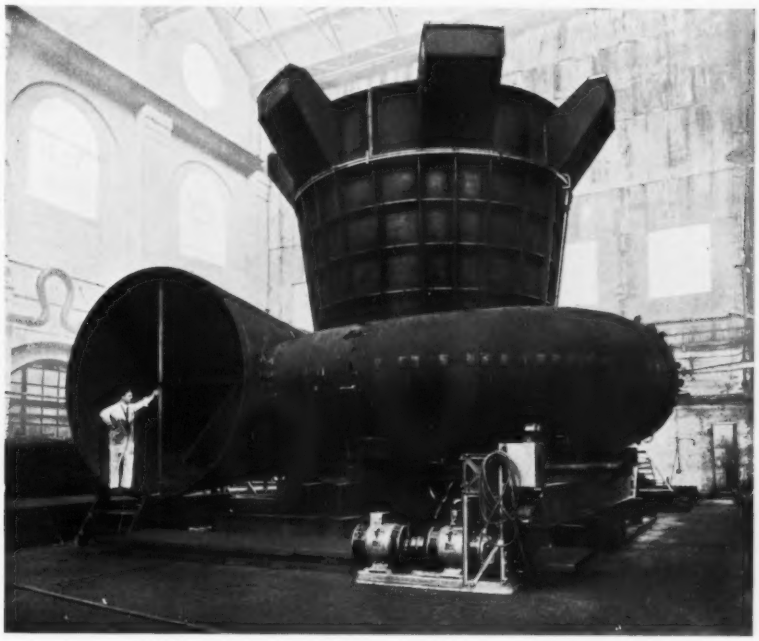


Fig. 61.—Spiral Casing and Generator Support Assembly.

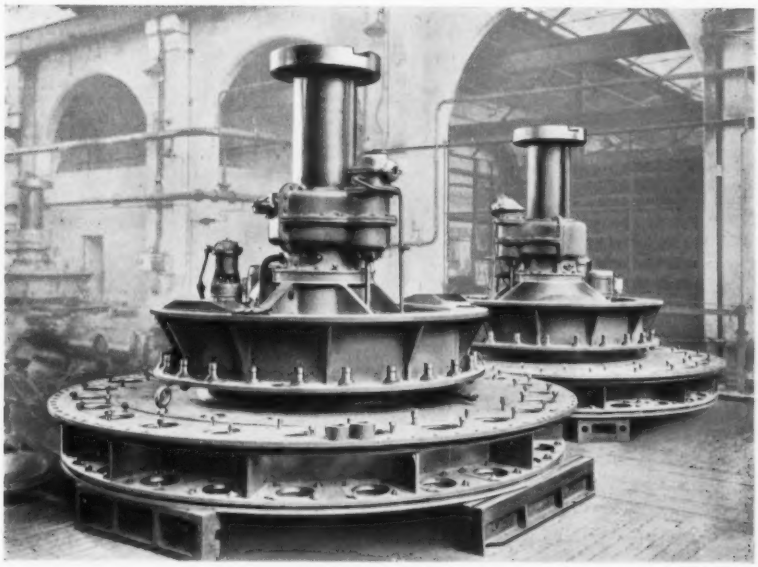


Fig. 62.—Turbine Top Cover Assembly.

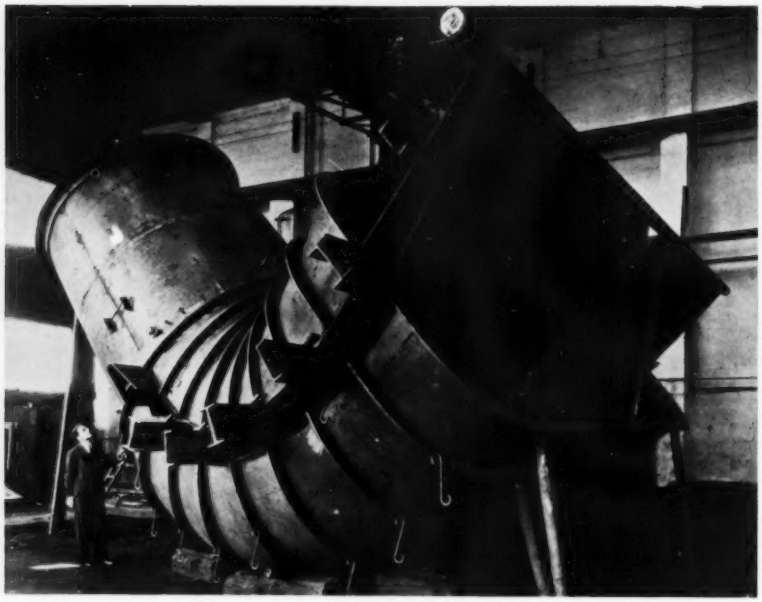


Fig. 63.—Draft Tube Assembly.

The torque is no longer transmitted by the bolts, but by one single, amply designed, diametrical key, traversed and kept in position by two of the bolts, and made to jig to fit the coupling halves with a slight interference fit. This jig contains an exact centring, fitting each coupling so that the relative positions between centre spigot and keyways are made to correspond exactly. The number of elements which have to be made to fit exactly is thus reduced to two instead of a multiple of this number. This type of coupling, which is illustrated in Fig. 58, is easy to manufacture and handle, and obviates reamering in the field.

The Development of Welded Structures.

The necessity of reducing both time of delivery and weight has led to the development of composite welded structures on a considerable scale. Not only pipes, but spiral casings, turbine main covers, valves, bearing housings, draught tubes, barrels, rotors, and even runner wheels are now fabricated by electric welding with carefully selected electrodes and welding plant. Experience has shown that most of these fabricated structures are not only lighter and cheaper, but much better than many castings, inasmuch as they are free from blowholes, cracks, and other defects always to be expected, especially in complicated steel castings. The considerable progress made in this field can best be illustrated by examples of the most important pieces, most of which have been in continuous and successful operation for a considerable time.

Spiral Casing.

The correct logarithmic form for an intake vortex is made up in lobster-back shape, which is obtained by butt-welding the composite parts joining the gradually diminishing sections, and

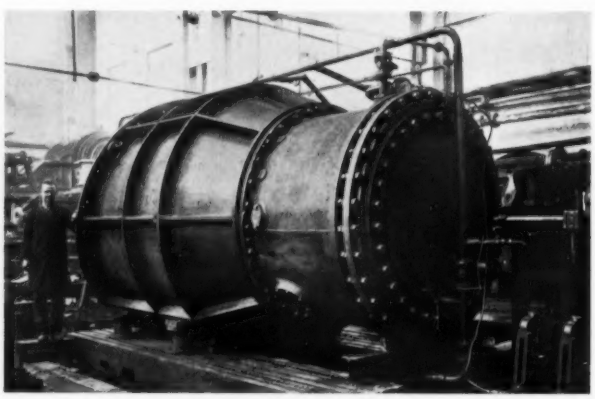


Fig. 64.—Relief Valve on test at works.

double lap-welding on to the speed ring or centre portion which surrounds the guide apparatus. The longitudinal joints, tail ends, and manhole joints, which are exposed to the greatest stresses, are reinforced by multiple lap- or stitch-welding, which is in reality not absolutely indispensable, but is an additional safety factor for these joints and greatly increases their strength. The inside of the spiral casing is smooth and is on that account alone superior to the riveted or lap-welded casing as it presents a minimum of resistance and disturbance to the flowing water forming the intake vortex. There has been no case of failure under pressure or distortion in service with this type of casing.

Immediately above the spiral casing can be seen in Fig. 61 the barrel or generator support. This barrel is also entirely welded and while remarkably light is of considerable strength. It is made in several sections for transport and contains passages or ducts for inspection.

The turbine cover also formerly presented difficulties in manufacture, both of the pattern and of faultless castings. It is made up of top and bottom covers between which twenty-four or more different compartments must be made to correspond exactly with the equal divisions of the guide apparatus of the distributing gear. It can now be made comparatively easily and rapidly by welding. Such a cover (Fig. 62) presents an almost perfect homogeneity apart from being much lighter than a steel casting.

The fabrication by welding of draught tubes is a problem which has much in common with the fabrication of spiral casings. The correct transition from the circular to the parabolic exit form at the bottom is of considerable importance in effecting a smooth and efficient transformation of the kinetic energy contained in the flowing water as it leaves the runner, into negative pressure. Welding enables the draught tube to be made perfectly smooth inside and assists the correct and expeditious building of the foundations (see Fig. 63).

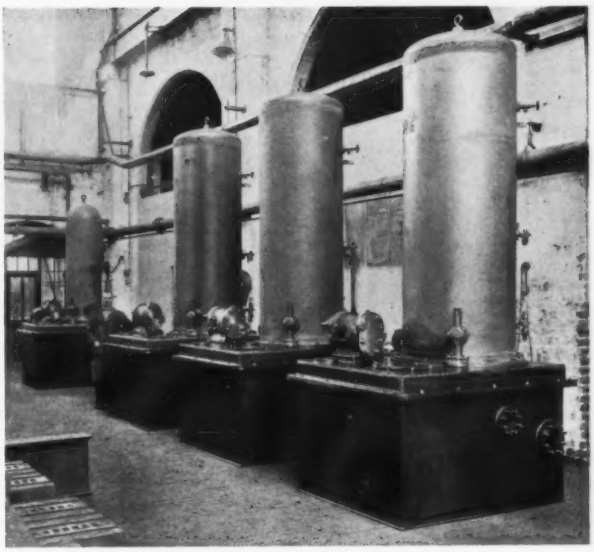


Fig. 65.—Governor Oil Pressure Pumping Sets and Receivers.

The relief and bypass valves lend themselves admirably to fabrication by welding. The entire outer body, dismantling ring, and discharge hood are welded. The valve piston is of phosphor-bronze and glides on stainless steel strips on the inner body. This relief valve, illustrated in Fig. 64, discharges up to 30 tons of water per second under 150 feet pressure and is much stronger and more homogeneous than a cast valve. The control and operating piping is connected to it by means of adequately reinforced double lap-welded joints.

The success obtained so far with welding is due to careful design, continuous tests of the strength and homogeneity of the welds, and minute supervision and control of the welding material and the welders.

Stainless Steel Protective Welding.

Great care is taken to avoid the danger of cavitation by a judicious choice of specific speeds and static and dynamic suction heads. The continuous increase of the specific speed of hydraulic turbines with a view to reducing first costs, however, coupled with the unavoidable deviation from the correct angle of the runner blades at part load, and change of pressure, make it

Fig. 66.—Turbine Runner showing stainless steel protection.

imperative to provide against the danger of cavitation which is so disastrous to both the life and efficiency of the runner wheels and turbines.

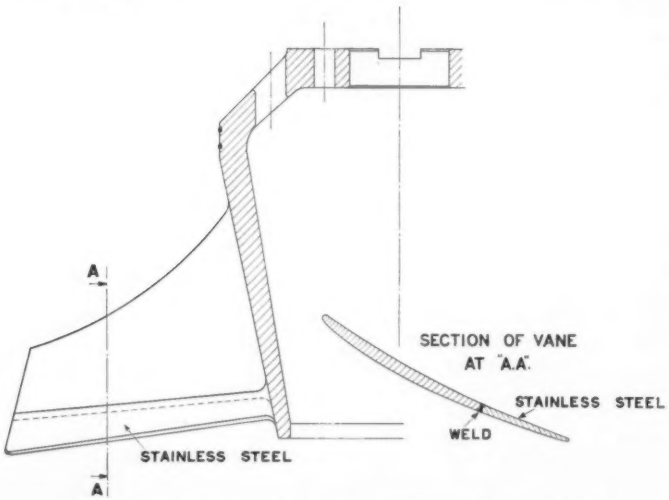


Fig. 67.—Showing stainless steel protection of Propeller Wheel.

The continuous increase of pressure utilized in both reaction and propeller wheels causes the water to traverse and leave the runner canals under far more severe conditions than those prevailing a few years ago. Cavitation is by no means restricted to water turbines running under certain exacting conditions, but is also in evidence in the case of ship propellers.

Protective measures must include the most careful design of the turbine runner blading and the choice of the best possible material. It has been found that "turbadium" bronze, specially created for service in ship propellers, as well as certain brands of stainless steel, show excellent cavitation-resisting properties. Unfortunately the price of these alloys is extremely high

and the production of sound castings, both in "turbadium" bronze and stainless steel, is attended with very considerable difficulties. Runner wheels work under considerable stresses and the risk of casting defects would be a serious consideration. Complete wheels of these alloys would, however, be unnecessarily extravagant because only a small portion of the wheel, the position of which is well known from hydrodynamic consideration and lies at the outer and lower part of the blading, needs protection against cavitation.

The first solution of the problem, therefore, consists in providing the critical region of the blading with a protective layer of stainless stee applied by welding on to the cast steel basis metal after careful preparation of the latter (Fig. 66). With careful handling no modification of the properties of the stainless steel takes place and wheels treated in this way have shown neither cavitation nor other defect.

The second solution is to cast the runner with blades of restricted length and to add tail ends of appropriate length and shape in forged stainless steel so that the whole critical region is protected. The junction between the cast and stainless steel portions is electrically welded after preparation and finally ground and polished. Figs. 67 and 68 illustrate this method of protection. It has the same effect as the stainless steel coating though it is somewhat more expensive, but it avoids the large amount of grinding and polishing necessary with the first method.

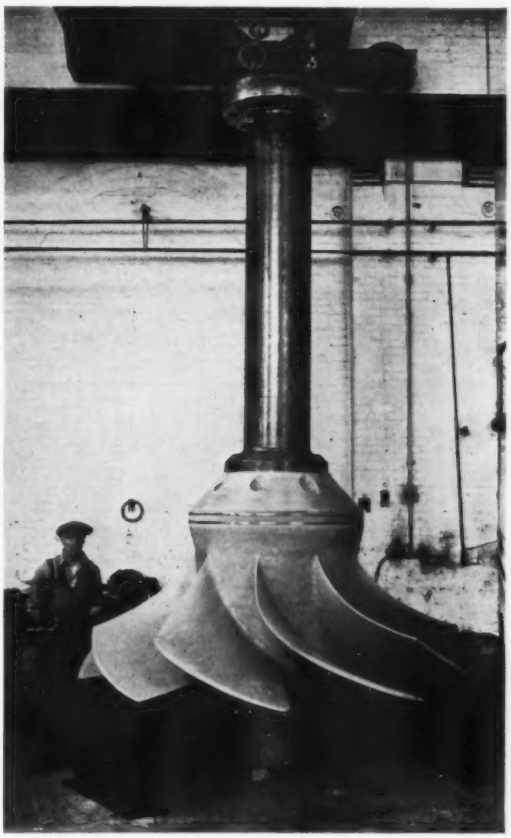


Fig. 68.—One of four 8,500 h.p. Propeller Wheels for Carsfad and Earlstoun Power Stations of the Galloway Water Power Company.

Conclusion.

In this paper the author has given a brief summary of the extent and importance of hydro-electric engineering and the progress, trend and lines of development of British practice. The purpose and application of a number of characteristic features for hydro-electric plant which the Company has successfully developed have been described.

It emphasises that hydro-electric engineering in this country is progressive and that it merits the attention and confidence of all those who are interested in the development of the vast water-power resources available in all parts of the world.

In previous issues of this Journal descriptions have appeared of hydro-electric plant manufactured by the Company, and the successful operation of these plants over many years is convincing proof that the confidence of British engineers and electric power companies in the Company's hydro-electric products is well founded.

Steam Turbine Research at Rugby Works.

By J. P. CHITTENDEN, M.I.Mech.E., A.M.I.E.E., Chief Engineer and Manager, Turbine Department.

Introduction.

The Rugby Works of the English Electric Company were founded in 1897 by Willans & Robinson, when expanding trade necessitated larger and more centralised premises for this historic firm than were provided by their works at Thames Ditton.

To the majority of steam engineers throughout the world the name of Peter Willans is familiar as one of the pioneers of steam engineering, and certainly as the first serious investigator into the systematic analysis of steam consumption trials. His name is perpetuated to the engineering world by the "Willans Line;" first propounded in his classic paper on "Steam Engine Trials," read before the Institution of Civil Engineers just prior to his death in 1893.

To his name may be coupled that of Captain H. Riall Sankey, a director of the original firm, who assisted in, and later continued, the work of Peter Willans. Captain Sankey was the first to realise the value of, and to apply in a practical manner to problems of steam engineering design, the entropy chart introduced by Macfarlane Gray, and later he further extended the value of this chart by the addition of the superheat lines.

Among other members of the Company who continued this work mention should be made of E. G. Izod, who was for some time Chief Engineer of the Rugby Works and was the originator of the impact testing machine which bears his name.

The spirit of research and original creation fostered by our pioneer founders has continued as an essential factor in the life of these works, and since the days of the Willans Central Valve Engine, which earned world-wide repute, many notable contributions to engineering achievement have left the Rugby Works as an outcome of this policy. To quote a few of the more outstanding examples, it may be mentioned that the first experimental Benson boiler, generating steam at the critical pressure (3,250 lbs. per square inch) was erected at the Rugby Works of the English Electric Company. This was the first occasion on which steam was generated at the critical pressure on a commercial scale beyond previous very small scale laboratory experiments. In connection with this super-pressure boiler an experimental turbine was built to operate at a steam pressure of 1,500 lbs. per square inch, from which much valuable data was established on the design and construction of turbine details for higher pressures and temperatures during the two years that the plant was run for experimental investigation.

Following this work to its practical realisation, the "English Electric" high-pressure installation at the Valley Road Power Station of the Bradford Corporation can be instanced. This plant operates at a boiler pressure of 1,100 lbs. per square inch, and was until comparatively recently the highest pressure installation in the country.

As a result of this, in 1927 the attention of the industry was drawn to the advantages of higher steam pressures and temperatures in a paper* read before the Institution of Electrical Engineers by A. H. Law and the author, and pioneer work in this direction by the English Electric Company established in this country the practicability of their application.

A direct outcome of the application of higher

* "*Higher Steam Pressures and Their Application to the Steam Turbine*"—*see I.E.E. Journal Vol. 66, No. 373.*

Fig. 1.—Steam Turbine Experimental and Test Shop.

initial steam pressures and temperatures is the superposed back pressure turbine applied to existing plant as a means of improving the thermal efficiency of low-pressure plant. The increasing use which is being made of this principle and the interest which is being shown in it by engineers in this country and abroad is added testimony to the Company's policy in carrying out its investigation and research on the behaviour and operation of steam turbines at high pressures and temperatures.

On parallel lines the Company has taken the initiative in the development of large capacity turbines running at 3,000 r.p.m. Probably one of the earliest 3,000 r.p.m. turbines to be built was a 375 kW. machine manufactured by the Company in 1906, which is still in commercial service to-day. As experience was gained outputs were gradually increased, and in 1915 the Company took the lead with a 5,000 kW. machine. Further improvements in design made possible continued advancement in outputs, and in 1921 the Company again came to the fore in building what was for a time the largest single-cylinder 3,000 r.p.m. turbine—a 15,600 kW. machine for Bolton Corporation. This machine is still in service, and has generated a total of 341,411,760 units to date. Following this, even still larger outputs were made possible by the introduction of their multi-cylinder design and the first 30,000 kW. turbine to be built in this country was supplied by the English Electric Company for the West Ham Corporation Electricity Department. At the present time the Company's Rugby Works are engaged in the production of two 40,000 kW. 3,000 r.p.m. machines, which are the largest to be supplied for operation in this country at this speed.

Achievement in the extraordinary in itself, however, is not necessarily a mark of integrity, neither does it hold much serious interest for the average engineer whose requirements lie in the direction of standardised construction and reliable operation, but it is well to remember that such achievement can come only as the direct result of very careful attention to the seemingly minor problems associated with plant of more conventional design.

For the investigation of such problems, the Company's Rugby Works are well equipped for the routine testing of materials to the final steaming test of turbines up to 50,000 kW. capacity, and for experimental and research work on the many components going to make up a machine.

Fig. 1 shows a general view of the turbine experimental and test shop in which are carried out routine tests on specimen bars of material destined for blading, and on test pieces from rotor forgings, castings, etc. This department is also equipped for testing and setting governors, each of which is individually tested over its full working range before being fitted to its turbine.

In addition to these tests, which are essentially of a routine nature, special tests of a searching character are made when introducing any new features in design, or when the complexity of factors involved renders mathematical treatment difficult. For example, before using a new blade form, a new type of root fixing, or a blade of exceptional length, tests are carried out on specimens of the new design to investigate efficiency of steam flow, mechanical strength under reproduced conditions of centrifugal force, natural period of vibration, strength under combined centrifugal force and bending loads, and, in the case of separately mounted discs, vibration tests are carried out on a completely bladed disc to ascertain the natural periods of vibration and so ensure that these periods of vibration are sufficiently far removed from the running speed for safe operation.

Similarly, any problems arising out of the running of machines on site are dealt with analytically, and in such instances much valuable work is done in the chemical and metallurgical laboratories at Rugby Works. Such problems as the metallurgical examination of forgings during machining processes and photomicrography of materials, water analysis as bearing on condenser tube corrosion, feed water analysis, oil tests, etc., all come within the scope of the equipment and staff of the Research Section of the Steam Turbine Department.

Steam Research.

A considerable amount of attention has been directed during recent years to the study of flow, both liquid and gaseous, and the results of many investigators tend always to show that the problem is by no means a simple one and that the prediction of results is fraught with many pitfalls even with the assistance of the results of scale experiments. The familiar difficulties associated with air and water flow are present in the case of steam, with the added complication that experimental observation is hampered by the inconvenience of high temperatures. Various means have been devised to surmount the obstacles to experimental investigation, but even so the application of experimental results to practical design cannot be reduced to a problem of simple substitution.

It is unavoidable, therefore, that the turbine designer is thrown back to the final full scale turbine for the ultimate proof of his design. Experiments on such a scale, however, are costly, and radical alterations to the steam path are therefore introduced with caution and by a process of elimination on scale experiments gradually approaching the full-size machine.

The general procedure when investigating proposed modifications is to build up a representative model of the detail in question suitable for experimental examination, and submit this to a detailed analysis under conditions as closely approximating normal operating conditions as possible. When this has been developed to the limit the newly designed detail is applied to a small experimental turbine kept expressly for this purpose. If necessary, variations of the design are tested in this machine under identical conditions and the best is selected from the results obtained.

If the anticipated improvement is realised, the design is applied to the full-scale machine and the investigation is followed up to finality by taking exhaustive tests centring round the newly introduced detail under actual full load operating conditions.

Fig. 2 shows the experimental turbine used in the development of new designs. This machine is so constructed that it can be readily adapted to take new diaphragms or rotor discs for such trials, and from a study of the pressure and temperature distribution through the stages, and from the results of steam consumption tests, which can be carried out up to a flow of 30,000 to 40,000 lbs. per hour, the effect of small changes can be accurately deduced.

The turbine is designed to carry four or more stages of expansion, either of the impulse or reaction type, and exhausts into a surface condenser from which the condensate is led to weigh tanks for measurement. When it is desired to reproduce back pressure conditions at the turbine, the exhaust steam can be throttled by means of a valve provided in the steam line between turbine and condenser for this purpose. This arrangement permits back pressure operation while at the same time retaining the accuracy of the method of measuring the steam flow by weighing the condensate.

It will be appreciated that at the present state of steam turbine design, where efficiencies closely approach the theoretical maximum, any further noticeable improvement can be realised only by an accumulation of small gains at different points in the machine. Therefore, it is important, when conducting experimental tests of the nature described, that the accuracy of measurements should be of a very high order. With this object

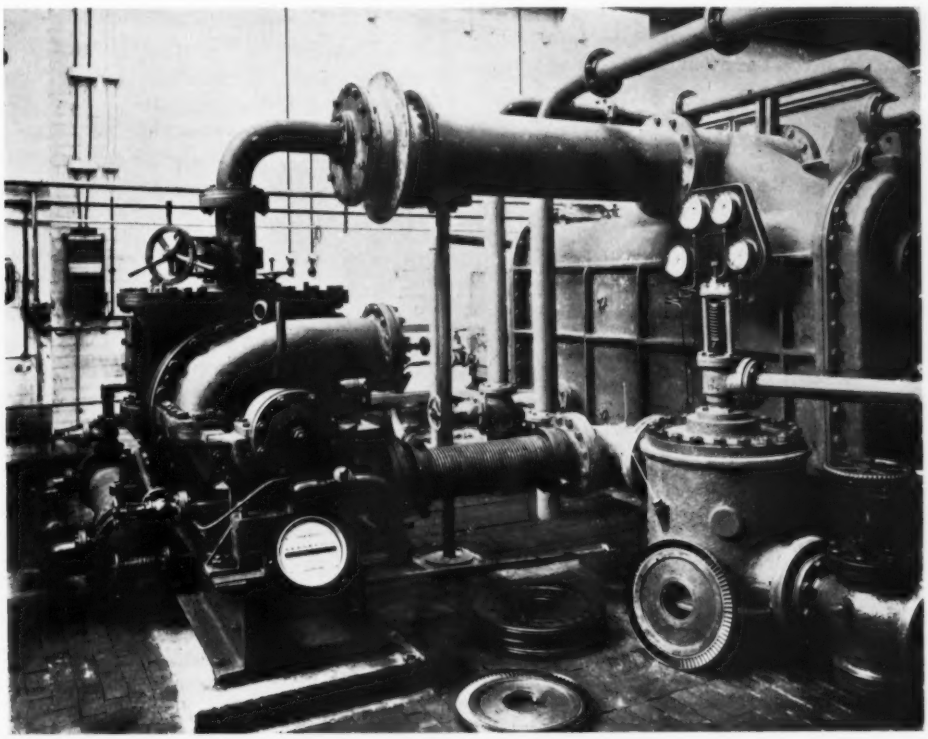


Fig. 2.—Experimental Turbine for development tests.

in view, the design of the experimental turbine is as simple as possible, all external loads such as the oil pump and governor are eliminated from the turbine itself, and the only losses are those due to windage and bearing friction. The whole of the available power is measured by a water brake coupled to the turbine shaft.

Among the problems which have been investigated along these lines are, for instance, the design of blade sections, influence of nozzle chamfer on steam efflux angle, effect of blade width, design of shaft glands for minimum leakage, questions affecting blade lap and clearances, steam joints, water erosion of blading, a number of which are described in detail later in this article.

In addition to the purely thermodynamic side of steam research work undertaken at the works, there follows as a natural consequence a certain amount of work in correlating the desirable optimum design with the nearest practicable approach as governed by manufacturing possibilities. As an example, the introduction of built-up steel nozzles by the English Electric Company some few years ago was designed to overcome the shortcomings in the region of high pressures and temperatures of the more usual construction of cast iron diaphragms having steel nozzle plates cast into the rim. With the growing tendency towards high temperatures cast iron became no longer suitable, and similarly as pressures rose and consequently the steam density, the smaller cross section of the steam path which resulted made the efficiency of flow very sensitive to small inaccuracies and irregularities in the passages. The built-up nozzle eliminated these disadvantages by enabling the use of steel throughout and also, by virtue of the fact that the boundary faces of the nozzles are machined over their whole area, a high degree of accuracy in the resulting passages is obtained.

The cast-in type of nozzle is, however, still retained for use in the lower pressure stages where the volume of steam is large and the effect of small surface inaccuracies is therefore of little account. The precise effect of these small variations is, however, known, and by keeping them within given limits, investigation has proved that they are of no consequence in the final performance of diaphragms used within a certain range of conditions.

For the detailed study of steam flow through blades or nozzles, exploratory tests through enlarged wooden sections may be carried out by using compressed air as a means of determining the flow distribution and ensuring the absence of eddies.

Thus, when conditions necessitate the introduction of a new blade or nozzle form, the steam path may be explored in this manner and the section developed according to the findings. This section is then made the basis for the design of a blade suitable for manufacture, and a batch of blades to this design is tested in a blade-testing apparatus, which gives a relative indication of the blade efficiency as compared with a known standard of reference. If this test shows that the designed improvement is being realised from the point of view of steam flow, a further test is carried out under actual operating conditions in the experimental turbine previously described, where a more exacting investigation can be made, than is provided by the static blade test.

A similar procedure is adopted when investigating the effects of other details, and in many cases comparative tests have been taken on full-size turbines operating under normal full load conditions in order to establish the evidence obtained experimentally. The following pages describe a few of the problems dealt with in this manner and give briefly an indication of the objects involved rather than attempting to describe in detail the methods employed.

As has already been mentioned, investigation of the effects of blade lap, width and nozzle chamfer has been carried out. The object of blade lap is best described perhaps by means of Fig. 3. Due to the fact that a steam jet leaving a nozzle has a natural tendency to spread, it becomes necessary to provide an opening in the blade receiving the steam, which is larger than the nozzle from which the steam issues. If the difference between the two openings, or lap, is too great, turbulence will occur in the boundary

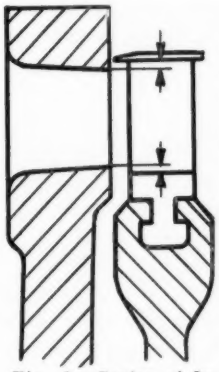


Fig. 3.—Section of Impulse Stage, illustrating blade lap.

regions with consequent loss of efficiency. A similar situation arises in connection with the flaring of blades to give the required blade exit area. If the blade width be made too narrow the angle of flare will be steep, resulting in an abrupt change in direction of the steam path boundary, with a consequent turbulent flow in this region. The choice of blade width is governed also by other considerations. The fundamental function of turbine blading is performed by virtue of the fact that the blades change the direction of the steam passing through them. If this change of direction takes place abruptly, turbulence and eddying will occur at the expense of efficiency, and thus it becomes necessary to select a blade width that will allow of the steam being turned through a sufficiently large radius to avoid undue disturbances in the flow. On the other hand, considerations of overall length, cost, etc., tend to limit the increase of blade widths, and the optimum value must therefore be sought by experiment.

Another aspect of the steam path which has an important bearing on the efficiency of turbine operation is the influence of nozzle edge chamfer on the steam efflux angle. Various degrees of chamfer were tried and it was found that not until an appreciable chamfer was used was there any reasonable approach to uniformity in the jet angle.

In order to ascertain the full significance of these effects, preliminary tests were carried out on the experimental turbine with different degrees of chamfer, and the best of these was used in comparative tests on two 1,000 kW. 3,000 r.p.m. turbines and repeated on a 5,000 kW. turbine of similar speed. These machines were first run with unchamfered nozzle discharge plates and later tested after the edges had been chamfered. These tests established conclusively that an appreciable improvement followed the use of chamfered discharge edges, and gave, too, a considerable amount of valuable information on the degree of chamfering necessary, together with the correct efflux angle to produce the best results.

The accumulated effect of minor changes has to be considered in relation to the turbine as a whole to determine whether results justify the elaborations and to conclude what are possible secondary effects which may not have been foreseen in considering the detail. For this purpose elaborate and searching tests are carried

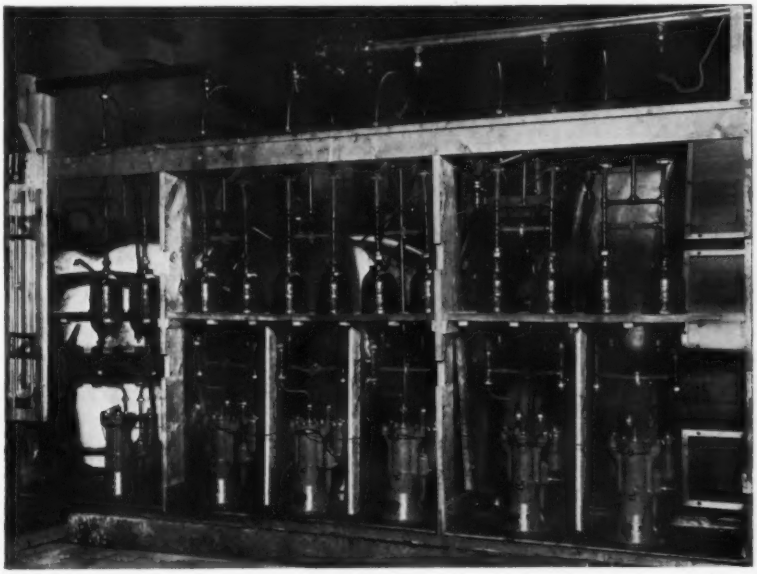


Fig. 4.—16,000 kW. "English Electric" Turbine, equipped with special test apparatus.

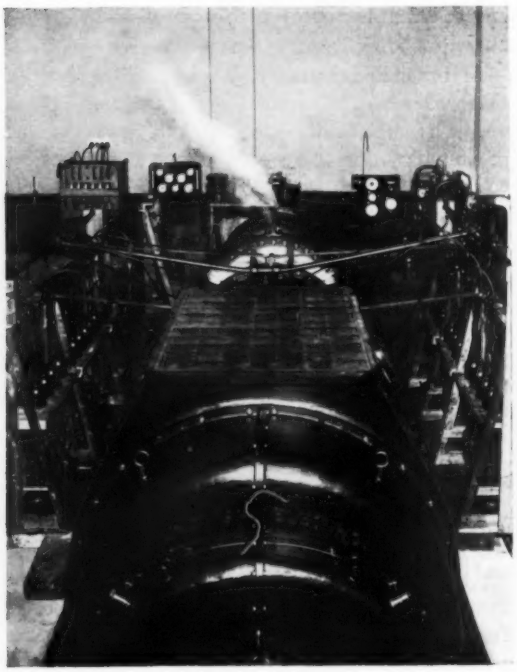


Fig. 5.—Special test apparatus on the low-pressure cylinder of a 12,500 kW. "English Electric" Turbine.

out on selected machines when running under load conditions. Such trials as are generally carried out on all turbines when completed yield useful information as to the overall performance of the machine, but their scope does not embrace the detailed analysis that is desired. A step-by-step investigation is required in order to ensure that improvements gained in one direction are not rendered nugatory by retrogressive steps or secondary effects in other directions.

Fig. 4 shows a 16,000 kW. two-cylinder "English Electric" turbine* equipped with special test apparatus for the investigation of pressure distribution throughout the turbine stages. Owing to the small differences of pressure existing across many of the stages, particular attention had to be paid to the method of measurement in order to secure accurate results. Both the high-pressure and low-pressure cylinders of this machine were explored under varying conditions of load and steam conditions, and the experiment formed one of a series taken on several different turbines to establish the effect of a revision in blading design.

Similar apparatus was used in another power station to investigate the effect of superheat or wetness as applied to the low-pressure turbine of a two-cylinder machine (see Fig. 5). In this case the double-flow low-pressure reaction turbine was disconnected from its high-pressure turbine and was supplied with steam through a reducing valve and desuperheater. The independent running of this part of the machine isolated from the usual governing gear, involved the arrangement of special safety devices for the control of steam to prevent over-speeding and for the electrical disconnection of the alternator from the station busbars in the event of an emergency. The inclusion of a reducing valve and desuperheater permitted tests being taken at varying steam pressures and permitted control of the quality of the steam supplied to the turbine.

These tests had as their object the investigation of the effects of wetness at the low-pressure end of turbines, a factor which was of prime importance both as regards operating efficiency and life of the blading, as with increased initial pressures and higher efficiencies the degree of wetness increased to such an extent that it became imperative to mitigate the ravages of water in the last stages of steam turbines. A considerable amount of data was obtained from these tests relating to the behaviour of steam through the low-pressure portion of the turbine at various inlet temperatures, together with degrees of wetness at the last stages, and, incidentally, data was also secured as to the value of interstage reheating.

In considering the problem of water-cutting of blading, three methods of approach were adopted, viz., the influence of material, the degree of hardness of the inlet edge, and the abstraction of a portion of the water.

The question of material was first thoroughly investigated, and numerous types of non-ferrous and ferrous alloys were tried; finally, after tests extending over two years, it was conclusively proved that the composition of the material was

* *See English Electric Journal, October, 1928, Vol. IV, No. 5.*

of little importance whereas the hardness of the water-stricken edge exercised a profound influence on the rate of erosion.

The results of one of these tests is shown in Fig. 6. Five blades of various steel alloys, together with one blade having a hardened steel edge, were inserted in the same stage of expansion of a machine and run for some 18 months at a high peripheral speed (about 1,000 ft. per second on the outer diameter) and a degree of wetness of about 13 per cent.

The illustration shows the five blades at the end of the test period, and clearly indicates the relative degrees of erosion on the different samples corresponding to the following lettering of blades :—

(A) Stainless iron, 0.1 per cent. carbon content.

(B) Stainless iron, 0.15 per cent. carbon content.

(C) 5 per cent. nickel steel.

(D) Mild steel with upper part of inlet edge shielded by hardened steel.

(E) Special austenitic steel.

Unlettered blade—mild steel.

Further experiments established that for effective resistance to erosion the degree of hardness of the blade inlet edge should be equivalent to a Brinell number of 600, and the method now adopted by the English Electric Company to accomplish this is to braze shields of 18 per cent. tungsten steel to the inlet edges of all blading running at high peripheral speeds and working in steam having a large degree of wetness. By careful control of the brazing temperature the tungsten steel shields are hardened to the requisite degree in cooling.

The third method tried for the reduction of water-cutting at the last stages of steam turbines

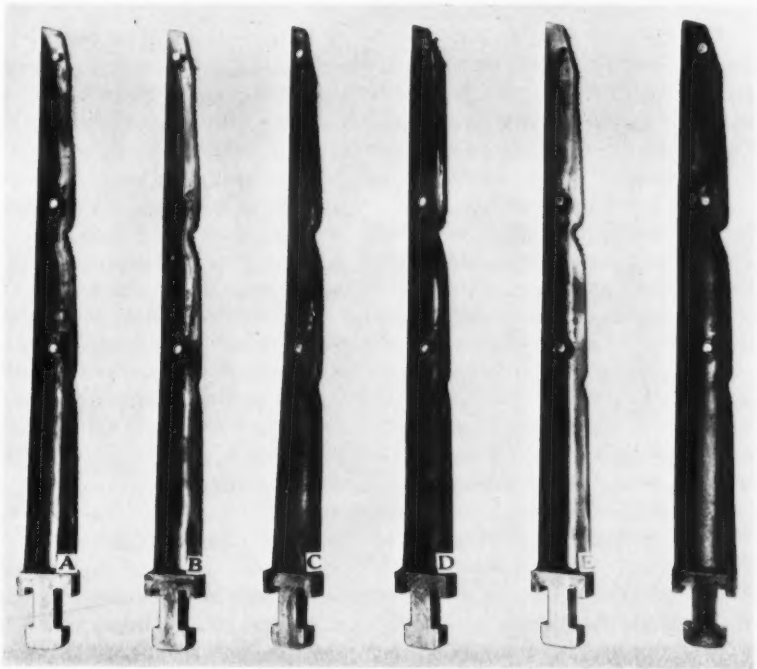


Fig. 6.—Erosion test—effects on blades of different material after 18 months' service.

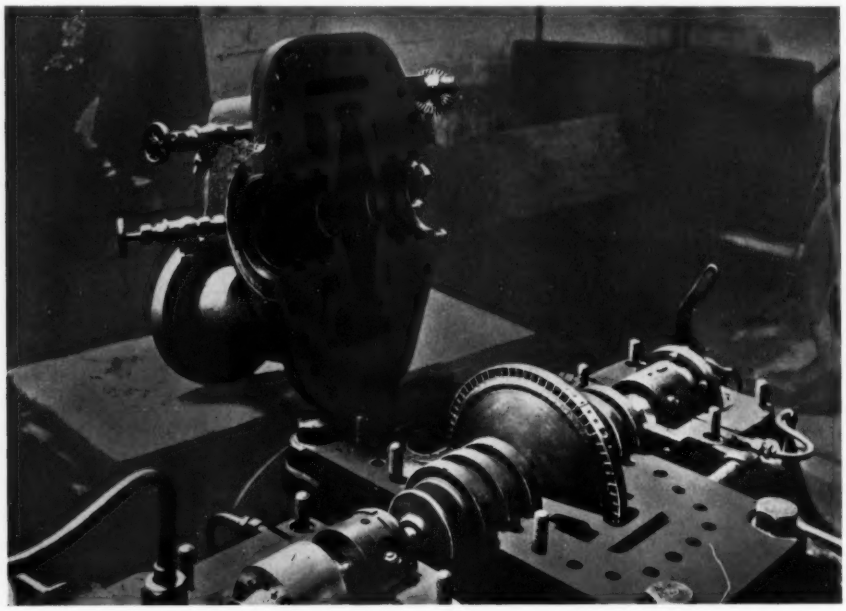


Fig. 7.—Experimental Turbine designed to operate in conjunction with 3,250 lbs. per square inch Benson boiler.

was by the extraction of water from the machine at these stages.

A large number of experiments were carried out with various types of water-catching grooves and baffles and a satisfactory design was evolved, which, together with special control valves, enabled an appreciable amount of water to be withdrawn, the result being that not only is the amount of erosion reduced, but a gain is effected in the efficiency of the stages concerned. These devices are now adopted on all " English Electric " turbines where the wetness and blade speeds are such as to warrant its inclusion.

Mention was made in the introduction to this article of work carried out by the English Electric Company on high-pressure steam research, and a more detailed account of the experimental plant installed at Rugby for this purpose may be of interest.

In this connection the first experimental Benson boiler was installed at the Company's Rugby Works in 1922, to investigate the possibilities of generating steam at the critical pressure and temperature. To operate in conjunction with this a 250 kW. high-pressure back-pressure turbine was built for experimental investigation into the problems arising from the use and behaviour of steam at very high pressures and temperatures.

This was the first occasion on which steam had been generated at the critical condition on a commercial scale, and marked the beginning of a period of great activity in high-pressure high-temperature steam research.

At the critical pressure, which is about 3,250 lbs. per square inch, the specific volume of steam is the same as that of water. Consequently, the formation of steam takes place quietly and without ebullition ; moreover, the equality in specific volume of steam and water entails no change in velocity with change in state, and therefore obviates the necessity for steam drums common to boilers of lower pressure.

Saturated steam at such pressures cannot, of course, well be employed at the inlet to turbines on account of the high degree of wetness developed during expansion with its attendant adverse effect on efficiency, etc. Consequently, in order to bring the turbine experiments within the scope

of practical operation for a machine of this size, the steam, generated at 3,250 lbs. per square inch, 742 deg. F. total temperature, was throttled down to a pressure of 1,500 lbs. per square inch and superheated to a total temperature of about 850 deg. F.

The turbine illustrated in Fig. 7 comprised a single-stage impulse wheel of 9.65 in. mean diameter running at a blade tip velocity of 1,100 ft./sec. and exhausting at 200 lbs. per square inch. The disc carrying the blades was forged solid with the flexible shaft running at 25,000 r.p.m., which drove a water brake at 6,000 r.p.m. through reduction gearing.

The blade root attachment was of the side-inserted type, which is easily removed, facilitating its replacement for tests on various blade forms. Shaft glands were of the carbon labyrinth type with end-tightening adjustment.

As in the case of the low-pressure experimental turbine previously mentioned, all auxiliary loads incidental to the turbine, such as oil pumps and speed governor drives, were omitted from the shaft and, where necessary, were provided for independently in order that a true value of the turbine's output could be arrived at with the greatest possible accuracy.

In connection with the actual tests carried out on this machine, exhaustive investigation was made over a period of two years into questions affecting reducing valve and throttle valve design for high-pressure high-temperature operation, and steam joints. Concerning the turbine itself, a considerable amount of valuable information was obtained on the subject of high-pressure shaft glands, disc friction and windage losses in steam at high density. A series of tests taken over a wide range of speeds and at different flow quantities established important data on the design and operation of blading and nozzles in high-pressure steam.

Mechanical Research.

Important problems involved in turbine design and construction are those associated with the blading and the rotor carrying it, as not only must the desired thermal efficiency be obtained, but mechanical stability under the combined effects of pressure, temperature, centrifugal and boundary loading at all conditions of operation is necessary.

As regards mechanical stability, many of the problems concerned can be resolved into comparatively simple cases readily amenable to calculation and giving results which can be relied upon with safety. There are circumstances, however, where mathematical treatment alone cannot be regarded with satisfaction, chiefly because of the complexity of factors involved. In such cases, elaborate technical investigation is undertaken to ensure complete stability under all running conditions.

Probably one of the least easily predictable problems from the mathematical standpoint is that of certain types of vibration. As in most types of high-speed machines, many parts of a turbine can, under given circumstances, vibrate in a number of ways. Some of them are, however, undesirable from the point of view of smooth running rather than dangerous. For example, insufficient provision for free relative expansion of the casing and rotor, particularly in the case of a multi-stage single-cylinder machine where the whole of the temperature range is confined to one casing, may well cause the turbine to run roughly, and this is, of course, a matter which is now especially allowed for.

There are, however, other types of vibration arising in the different parts of the machine which may lead to serious danger, and, indeed, to which, more particularly in the early days of the steam turbine, the wreckage of more than one machine can be traced, details of which are referred to later.

It is, of course, clear that it is essential that the rotor shall be in accurate balance, both statically and dynamically, and so designed that the main criticals and sub-criticals are well removed from the running speed.

Static balance is comparatively easily obtained by supporting the rotor with its journals resting on carefully levelled knife edges and adding balance weights, which are later used as the basis of calculating the amount of metal to be removed, or for permanent balance weights to be added according to the type and design of the rotor.

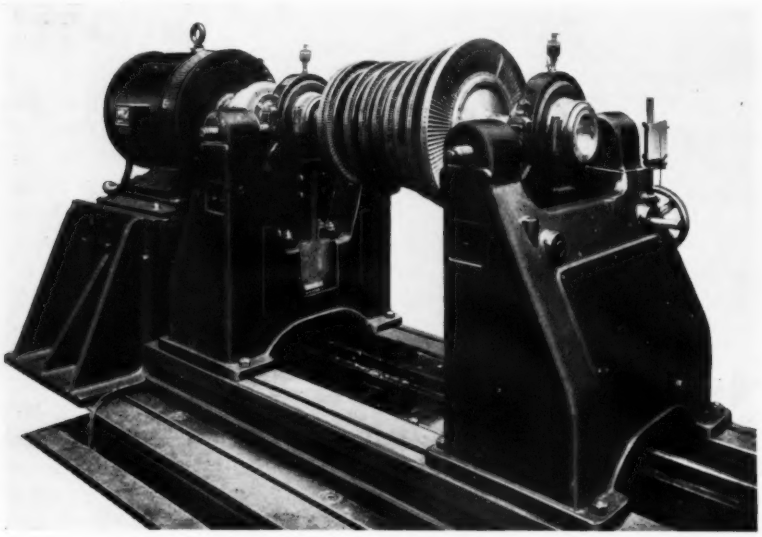


Fig. 8.—Dynamic Balancing Machine for small turbine rotors.

A rotor may be in perfect static balance, but it is still possible for it to be out of balance due to unbalanced centrifugal couples when rotating at speed. Dynamic balance is a rather more difficult operation and necessarily must be undertaken as the result of observations taken whilst the rotor is revolving. Developing from rather crude beginnings, the design of a balancing machine, as the result of a great deal of thought and ingenuity, has now reached a pitch where the balancing of the largest rotors is a matter of precision instead of the trial and error principles formerly used. Fig. 8 shows one of the Company's balancing machines as used for small turbine rotors, while Fig. 9 shows another more generally employed for large turbine rotors and alternator rotors. Both of these machines enable dynamic balancing to be carried out with a high degree of accuracy.

With regard to critical speeds or shaft whirling, all rotors possess a series of characteristic speeds at which centrifugal forces tend to overcome the elastic resistance of the shaft to deflection. If a shaft is run near one of these speeds,

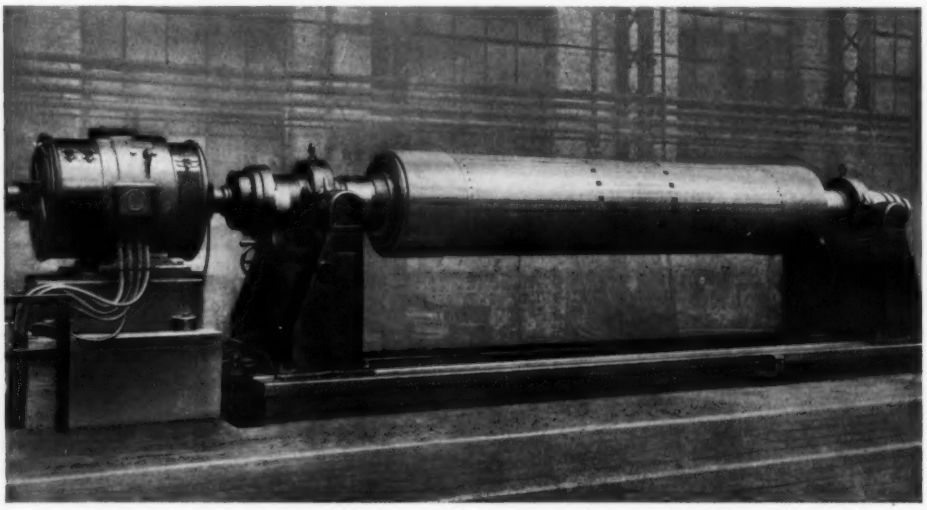


Fig. 9.—Dynamic Balancing Machine for large turbine and alternator rotors.

vibration will take place, and as the speed is made to approach the critical value, vibration becomes more pronounced until the critical speed is reached where, if the shaft is maintained at this speed for any length of time, the deflection becomes progressively greater until fracture occurs.

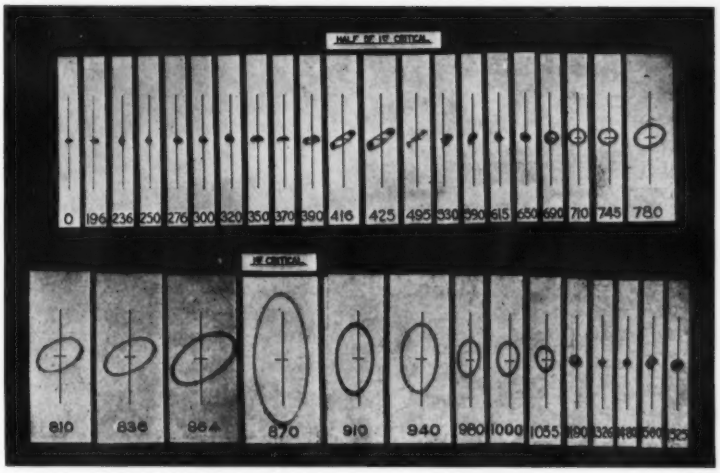


Fig. 10.—Record of vibrations of a model shaft at various speeds.

The first consideration, therefore, in designing a turbine rotor is to ensure that its whirling speeds are sufficiently removed from its operating speed so as to give rise to no trouble. Either the shaft is proportioned such that the lowest critical speed is well above the normal speed, or, if the size of shaft necessary to obtain this is such that consideration of disc hub stresses or the necessity of low interstage leakage losses in order to obtain high efficiency do not permit it, well below the running speed. In the latter case, care is taken to ensure that when running up to speed, the critical point is passed through as rapidly as possible.

The approximate determination of the lowest critical speed of a shaft—usually all that is necessary—is now a comparatively simple matter. A graphical method is generally employed.

Fig. 10 shows very clearly what takes place when a shaft is passing through its critical speed. It is a record of a model shaft with load distributed, obtained by means of a spot of light reflected from a mirror fixed to one end of the shaft. The diagrams indicate clearly how the shaft settles down to steady rotation in between successive critical points. The apparatus on which these diagrams were obtained formed part of an exhibit by the author before the Royal Society in 1924.

There is another form of shaft vibration or whipping which has in comparatively recent times been troublesome in some turbines, the cause of which was for a considerable time obscure. It is distinguishable from the true shaft critical speed in that it was found to occur over wide ranges of speed, the vibration or whirling frequency bearing little, if any, relation to the speed of revolution and being apparently unconnected with the dynamic balance of the rotor.

Considerable research and theoretical investigation have been carried out on this subject by, amongst others, Hummel, and it has been traced to oil action in journal bearings. It is now definitely established that the familiar wedge action of the oil in a well-lubricated journal bearing tends, in certain circumstances, to produce a " whirling motion " of the shaft in its clearance, travelling slowly round in the direction of rotation, but at a frequency of approximately half the shaft r.p.m., i.e., the journal makes one circle of the bearing in approximately two revolutions. This phenomenon results in a well-defined vibration (" Oil Resonance ") at twice the ordinary shaft critical speed, for the frequency of the journal would then be equal to the natural period of the shaft. Now, unlike the ordinary shaft critical speed, it appears that once vibration occurs at the " Oil Resonance " point, it is possible for the shaft to continue vibrating as the speed is increased, i.e., it fails to pass through this critical point. It has been suggested that the reason for this is that the greatly increased fric-

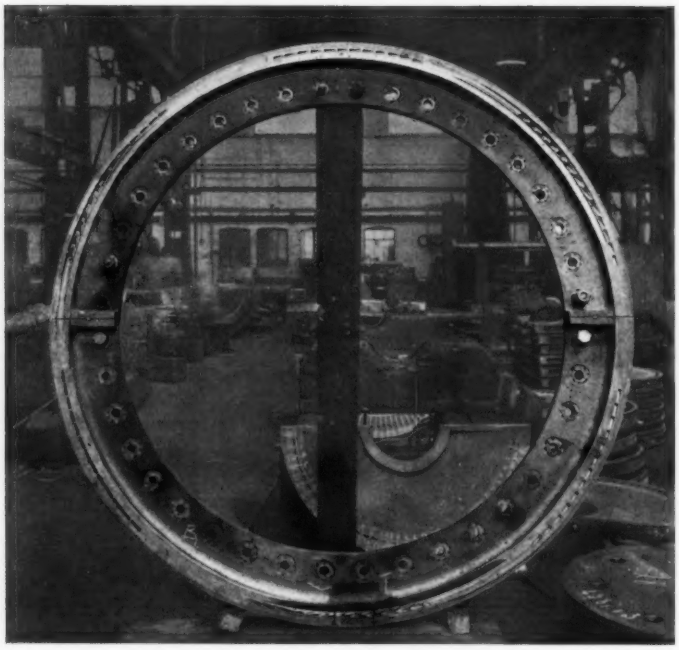


Fig. 11.—Nozzle Plate showing rubbing marks caused by a standing wave in a vibrating disc.

tional resistance caused by the whirling holds down the oil action to the resonant frequency so that it maintains the vibration even though the speed of rotation is increased considerably above the oil resonance speed. Further experiments by the Company showed that if a damping resistance were applied to a vibrating shaft just after the oil resonance point, the whirling could be prevented for a range of speed, but would often break out again at a higher speed. The effects of bearing clearance, length and loading were also investigated, and the main conclusions arrived at were that this form of vibration could be avoided by :—

(1) Confining operation to a speed below twice the first critical speed, though this is not usually practicable.
(2) Avoiding very light bearing loading pressures.
(3) Change in pressure of the oil supply.
(4) By reduction of bearing clearances.

A rather more serious type of vibration may occur in turbine discs and is thus a problem which more nearly concerns the design of turbines with large wheels of slender proportions. It was some time before the real cause of the trouble was traced to its source, and in some instances it was believed that the material was faulty. When, however, a disc apparently of perfectly sound material and stressed only to a moderate degree completely failed in service, and when replace discs to the same dimensions and specification behaved in a similar manner, some other root cause had to be found. Quite a number of ingenious theories were propounded before it was definitely established that axial vibration was taking place in the disc.

The comparatively light discs of large diameter which were at one time common in large single-cylinder machines running at 1,200 r.p.m. and 1,800 r.p.m. in the United States of America experienced more trouble than the smaller diameter discs which were more general in this country and on the Continent, though even here disc vibration was not unknown.

One of the earliest disc failures in this country was that of the first wheel of a 5,000 kW. 1,500 r.p.m. impulse machine. Under certain operating conditions this disc fouled the stationary parts of the machine with the result shown in Fig. 11. In this case the material was at first suspected, but when a second disc gave the same results, an investigation was made to determine whether the disc, when running, had a frequency which might coincide with any disturbing forces arising from the rotation of the set.

These early investigations were of a rather primitive nature and consisted chiefly in attempts to determine the natural period of vibration of the disc by means of a reed tachometer, while a light blow was given to the rim of the disc when held firmly at the bore. An effort was also made to

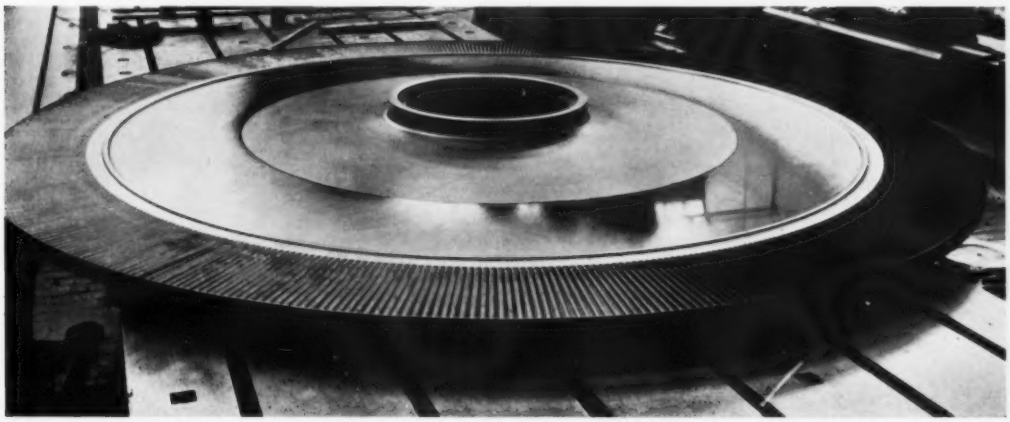


Fig. 12.—Static vibration tests showing a one nodal diameter vibration, using water as the indicating medium.

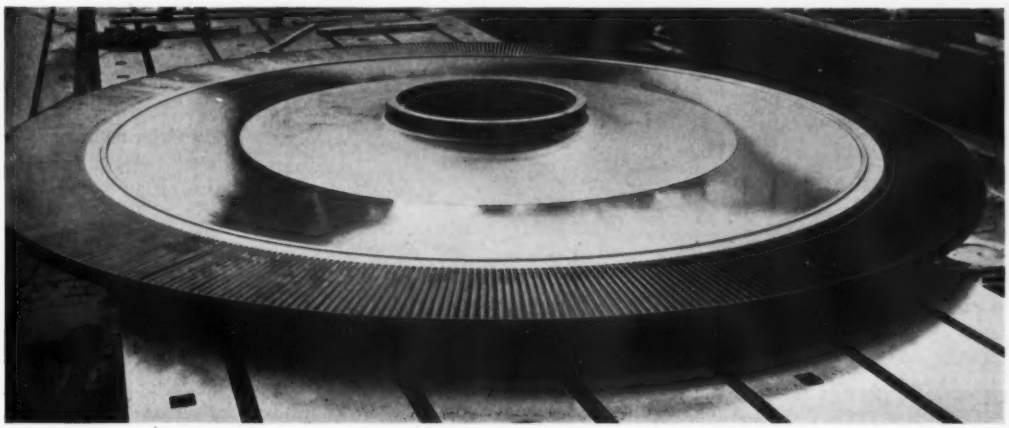


Fig. 13.—Static vibration tests showing a two nodal diameter vibration, using water as the indicating medium.

calculate the frequency based on the observed static deflection shape of the disc when lying horizontally and loaded at various points. These indicated that the disc probably possessed a natural frequency approximating to double the speed of rotation, but did not give any indication as to the manner in which the disc was vibrating, and methods yielding more definite results had to be devised.

The first step was to determine accurately the static vibration characteristics. The arrangement adopted by the Company was to bolt down rigidly the disc to a heavy face plate and excite vibrations in the disc by means of an electro-magnet fixed under the rim of the disc. The frequency of the impulses was varied by passing the magnetising current through a make-and-break device of the commutator type running on the shaft of a small variable-speed motor. The speed of the motor was adjusted until the disc vibrated in a manner indicated by the movement of a thin layer of sand or water placed on the disc. The usual form of vibration of turbine discs is in the nature of nodal diameters (i.e., the stationary

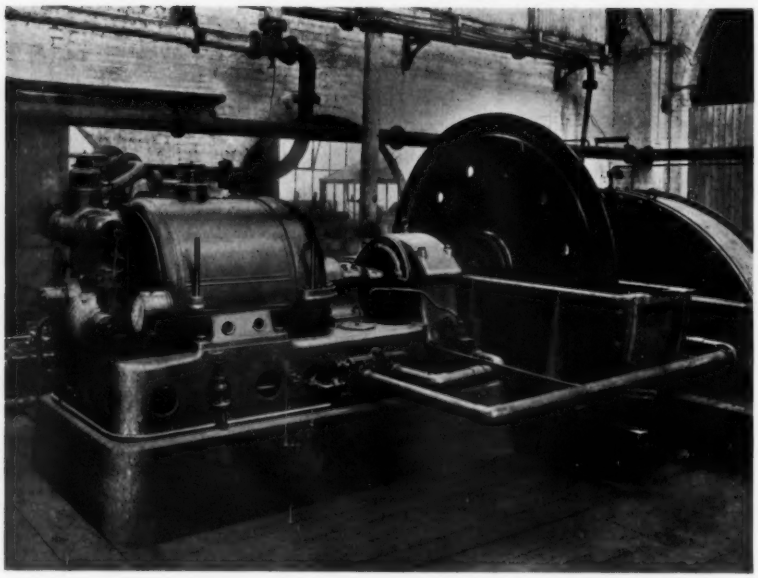


Fig. 14.—Disc Vibration Testing Machine with cover removed, showing a bladed disc in position.

points and the points of maximum displacement lie on the diameters of the disc either 1, 2, 3, 4, etc., in number). The discs may exhibit the phenomenon of cupping (or umbrella motion), but vibration on nodal circles and compound forms of vibration are not easily excited in turbine discs of normal design. Examples of such vibration figures taken on a disc of 96 in. mean diameter are shown in Figs. 12 and 13. These photographs show very clearly the nature of vibrations which take place. In the two examples illustrated, water was used as the indicating medium. At the nodal diameters where the deflection is zero the surface of the water remains clear, but elsewhere the vibration of the disc imparts a ruffled surface to the water.

It is, of course, obvious that if some certain theoretical method was available for calculating the vibration characteristics of discs, the risk of completely manufacturing a disc and then finding that it would be likely to vibrate in service would be avoided. The theoretical investigations were initiated by Dr. Stodola, and his fundamental expressions have been adapted by other investigators to suit certain types of discs. At their best, however, the methods are laborious, and, in addition, there are a number of factors whose influence on the period of vibration cannot accurately be estimated. Amongst these may be mentioned :—

(1) Method of support on shaft—whether disc is solid with shaft, shrunk on, or carried on mounting rings—and the degree of tightness of fit.
(2) The influence of blading and method of fixing.
(3) The influence of balance holes.
(4) The condition of internal stress throughout the web of the disc.
(5) Temperature.
(6) The influence of the speed of rotation.

If the effects of speed (i.e., of centrifugal force) were known with a reasonable degree of accuracy, some such static method as already described would suffice for the determination of the vibration characteristics of a disc. It would appear that an approximate relation (rotating frequency)^2 = (static frequency)^2 + K $\times$ R.P.M.^{2} holds good with fair accuracy, but unfortunately the factor K varies between 2 and 5 with the contour of the disc, blading load, and the number of nodal diameters.

For these reasons manufacturers have been compelled to rely on actual testing at speed of all discs where there is any possibility of vibration being set up, and for this purpose several have constructed elaborate testing machines in which discs can be examined for vibration under any conditions of speed, temperature, etc.

The machine built by the English Electric Company some years ago is shown in Fig. 14 ; the top half of the casing has been removed to

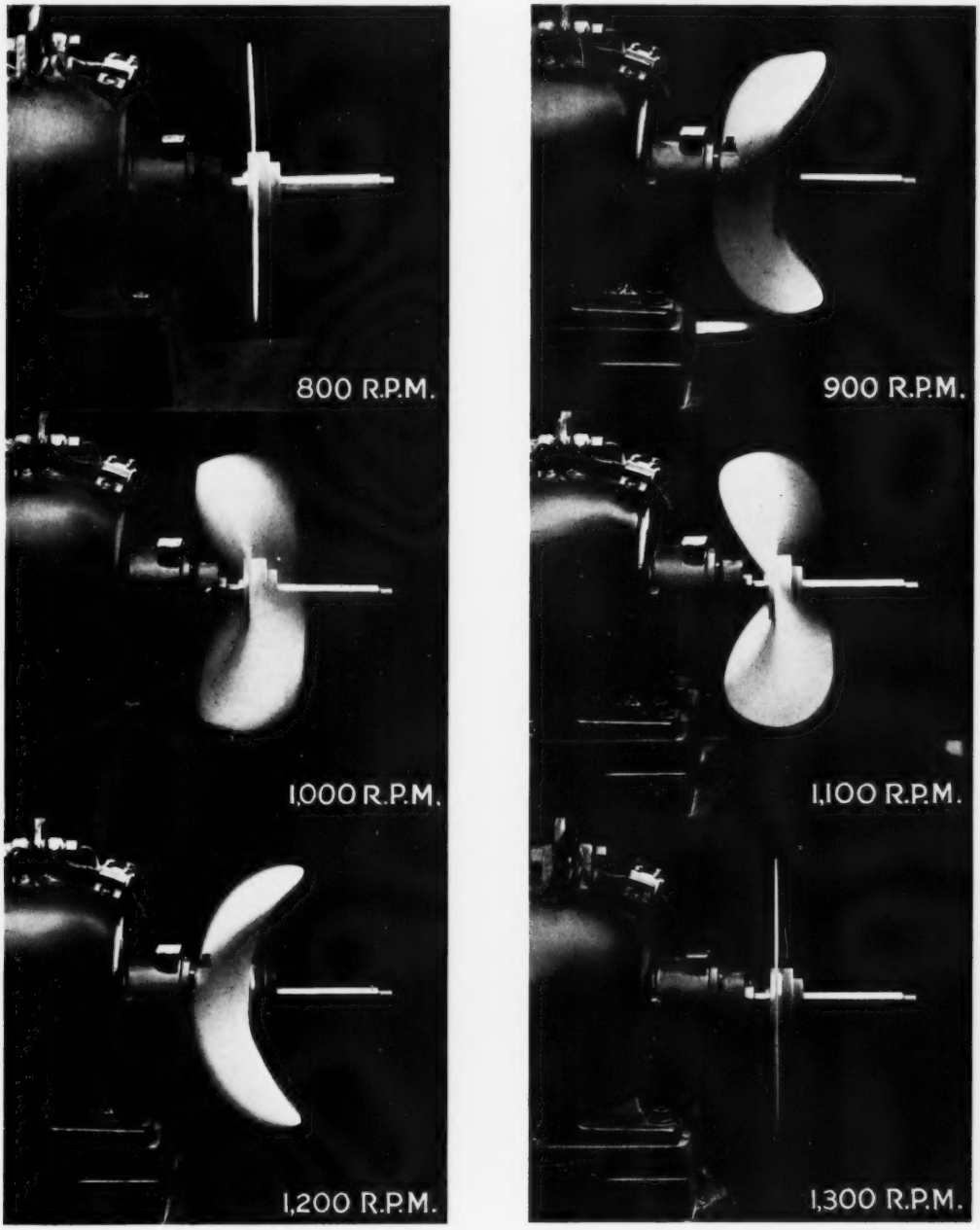


Fig. 15.—Illustration of a standing wave by means of a rotating rubber disc.

show the bladed disc in position. Vibrations of the disc relative to a stationary point are recorded by observing the variation in the current through an electro-magnet attached to the machine frame due to the changes in the air gap between magnet core and disc. Similarly, vibrations relative to a fixed point on the disc are recorded by the current variations in an electro-magnet attached to a stiff disc rotating with the bladed disc under test.

Campbell's fundamental discovery was that a rotating disc tends to develop trains of waves, corresponding to the various static nodal diameters, travelling round in the disc, either in the direction of motion or against it. Thus any point in the disc is continually passing through a succession of nodal points and points of maximum amplitude of vibration. It will be clear that such a vibration can be maintained by a pulsating force of the same frequency acting on the disc ; if such a force exists in a turbine, vibrations may be set up such that the resultant alternating stress superposed on the normal working stresses may exceed the fatigue limit, with ultimate fracture.

There is still, however, a more serious form of disc vibration. It is possible for a backward wave in the disc to travel round at the same rate as the disc is moving forward. These waves, although moving in the disc, appear stationary to an observer. In other words, the disc appears to be permanently deformed. The speeds of rotation at which this occurs are the critical speeds of the disc, corresponding to the various nodal vibrations (2, 3, 4, etc., nodal diameters).

The natural stiffness and the relatively small deflection of steel discs does not permit of the visual demonstration of their mode of vibration when running, but the nature of these vibrations can be very clearly shown in a flexible rubber disc which deflects in precisely the same manner but to an exaggerated degree. Fig. 15 shows a number of photographs of a rubber disc rotating at various speeds and vibrating at some of its critical speeds. These photographs were taken some years ago by the author on apparatus forming part of an exhibit before the Royal Society.

Such standing waves obviously do not require a pulsating force to excite them as in the case of the other resonant vibrations referred to above—a constant force at one point is all that is necessary. A force of only a few pounds is found to be sufficient to cause a disc to respond strongly to a whole series of critical speeds. It is in this characteristic that the danger to a turbine lies, for it hardly needs to be said that any irregularity in the nozzles, or more particularly the employment of partial admission in a diaphragm may result in a transverse steam force on the disc a few pounds greater at one part than at the others. If one of the disc's critical speeds lies close to the operating speed, it is safe to predict trouble. The lower nodal vibrations are usually of most importance since the vibration in these cases extends much further towards the centre of the disc and the amplitude of vibration tends to decrease with increasing number of nodes.

If the vibration is of sufficient magnitude to cause the disc to touch its diaphragm, the latter will show markings at a number of symmetrically placed points equal to the number of nodal diameters, whilst the whole rim of the disc will touch these points in turn. The rapid heating thus caused has been known to result in welding the rotating and stationary parts solidly together before the rim of the diaphragm was torn off, carried round with the disc and burst. In another case the breakdown may follow from a heavy out-of-balance due to blades being torn off or to fatigue starting from points of high local stress, such as balance holes.

It may happen that in spite of a complete knowledge of the vibration characteristics of previous similar discs, a disc is manufactured which, when tested, possesses a critical speed too near to normal running speed for safety. To avoid entirely replacing such a disc, recourse is made to tuning. By removing metal from a certain part of the disc profile or even from the blading, the critical speed in question may be reduced without unduly affecting the others. Thus, to influence a 2-nodal diameter vibration, metal is removed from near the hub, whilst a 6-nodal diameter vibration would probably be best corrected by removing or adding weight to the blading.

So much time and thought have been given to

the subject of disc vibration and conclusive data established that it can be said to be fairly well understood and very little trouble should be experienced from it in the future. It may possibly be mentioned here, however, that the modern multi-cylinder multi-stage machine running at 3,000 r.p.m., more particularly when of the solid rotor impulse reaction construction as typified by the Company's design, is inherently immune from disc vibration troubles. The lowest critical speed of its discs is very far removed above the normal running speed.

A disc is usually considered to be safe from vibration when the 2-nodal diameter critical speed is removed at least 15 per cent. and higher nodal diameter critical speeds 10 per cent. from operating speed. This margin is designed to allow, amongst others, for the effects of possible temperature variation, and the fact that vibration phenomena do not occur with precise accuracy at a definite speed, but may arise slightly above or below the speed corresponding to maximum resonance.

Closely allied to disc vibration is blade vibration. In fact, it is in some cases difficult to say where disc vibration ends and blade vibration begins, since, in a vibrating disc, blades and disc behave as a composite whole, and in the higher nodal vibrations any appreciable amplitude of vibration is often confined to the blading alone. Moreover, the length and weight of the blading has a very great influence on the natural frequency of the bladed disc. For example, from the discs of a single-cylinder 6,250 kW. 3,600 r.p.m. impulse turbine with discs of approximately 39 in. rim diameter, it was found that whilst a 1 in. blade reduced the static unbladed disc frequency at 2-nodal diameters to 83 per cent. of its value, an 11 in. blade reduced the figure to as low as 37 per cent.

However, it is possible for blades to vibrate independently of the disc, and various steps have to be taken, particularly in the case of long blades towards the exhaust end of a machine, to ensure that the natural frequency is left well removed from the normal operating speed or simple multiples thereof.

Fig. 16 shows the apparatus used for the determination of critical frequency and for fatigue testing of blades. In this case a D.C. energised electro-magnet, in conjunction with a simple make-and-break device mounted on the blades, causes sustained vibrations at the natural period.

The employment of shrouding, etc., has an important influence on the frequency of a row of blading, since its tendency is to cause blades to vibrate in groups rather than singly, and it also tends to change the mode of vibration from one with one node at the root, to one with a node at both tip and root. Both influences are in the direction of increased frequency. From this consideration it follows that the introduction of lacing wires at suitable intervals has a beneficial

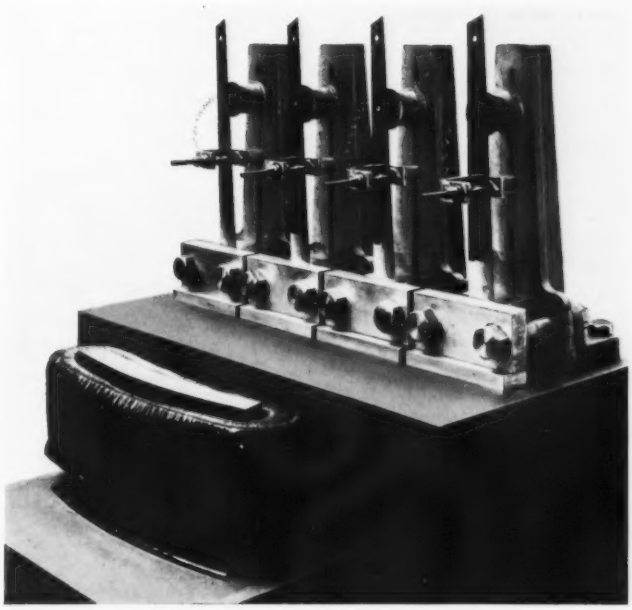


Fig. 16.—Blade Vibration Tester showing separately the electro-magnet used for group testing of laced blades.

effect, and indeed is essential in the case of long reaction blades of the radial clearance type. The number and size of such wires is, of course, only limited by considerations of efficiency and a compromise has to be effected.

For the testing of blades grouped together by shrouding or lacing wires an elongated electromagnet, shown in Fig. 16, is employed to induce vibrations in the group as a whole.

A development of these tests approximating more nearly to service conditions is obtained by the use of the apparatus shown in the left foreground of Fig. 1. Using this method, vibrations are induced in the blade, which is subject to a tensile load representing the centrifugal stress at a given point. In order that the attachment of the load shall have the minimum effect upon the freedom of vibration, the load is applied to the tip of the blade through a long horizontal wire.

It is clear, however, that none of these methods can reproduce faithfully the actual condition of stress and vibration under normal operating conditions. Reproduction, for instance, of centrifugal stress in blading under static conditions is an impossibility, but, nevertheless, by interpolation of the results of a number of tests by different methods, a method of approach may be arrived at by which, using a suitable safety margin, troubles arising from the vibrations described can be eliminated.

Although it is now safe to say that the causes of major vibrations are understood and therefore that troubles arising from them can be avoided by judicious design, there remain inevitably minor parasitic vibrations which, while not being dangerous in themselves, may, in conjunction with static or centrifugal stresses, lead to failure by fatigue.

As would be expected, fatigue effects of vibration make themselves evident by failure at the points of maximum stress. For example, turbine blades and their root attachments are subject to tensile, bending and shear stresses under the combined effects of steam pressure and centrifugal force ; they may also be subject to parasitic vibrations as a result of which fatigue failure is liable to occur if precautions are not taken in design to avoid points of local high stress. Now, although it is possible for normal design purposes to calculate the cross sectional area of metal to withstand known loads and by experience to allow for fatigue effects, the calculations are of necessity based on an assumed symmetrical and uniform distribution of stress throughout the material. That the stress distribution follows lines of flow analogous to flow in liquids is well known, but the actual prediction of the intensity of stress at any one point is a matter of conjecture without the use of special exploratory apparatus, such, for instance, as the Photo-Elastic apparatus based on principles outlined by Sir David Brewster and developed later by Professor Coker. Apparatus of this nature is employed by the Research Department of the Company's Rugby Works and the principle of its operation briefly is that specimens of celluloid or synthetic glass, when subjected to stress, become birefringent. Polarised light falling on the specimen is resolved into two components polarised in the directions of the principal stresses, and owing to the relative retardations between the two waves emergent from the specimen, interference occurs and colour effects are seen after the beam has passed through a crossed analyser, from which the stress distribution can be deduced and plotted or photographed for record purposes.

A considerable amount of useful information can be obtained by the use of this apparatus regarding the general conformation of plain members, but the exploration of stress distribution by the method is subject to limitations inasmuch that it can be applied only to thin laminæ representing the problem in two dimensions, and the effect of thickness cannot be ascertained. In the general case it becomes necessary, therefore, to carry out full scale tests to destruction in order to verify calculated designs of a complex nature or where the basis of calculation is in doubt.

As an example of this type of work, Fig. 17 illustrates the results of a series of tests carried out on blade fixings of different proportions in the same material in order to determine the optimum proportions of disc width to root section, a problem not readily amenable to calculation,

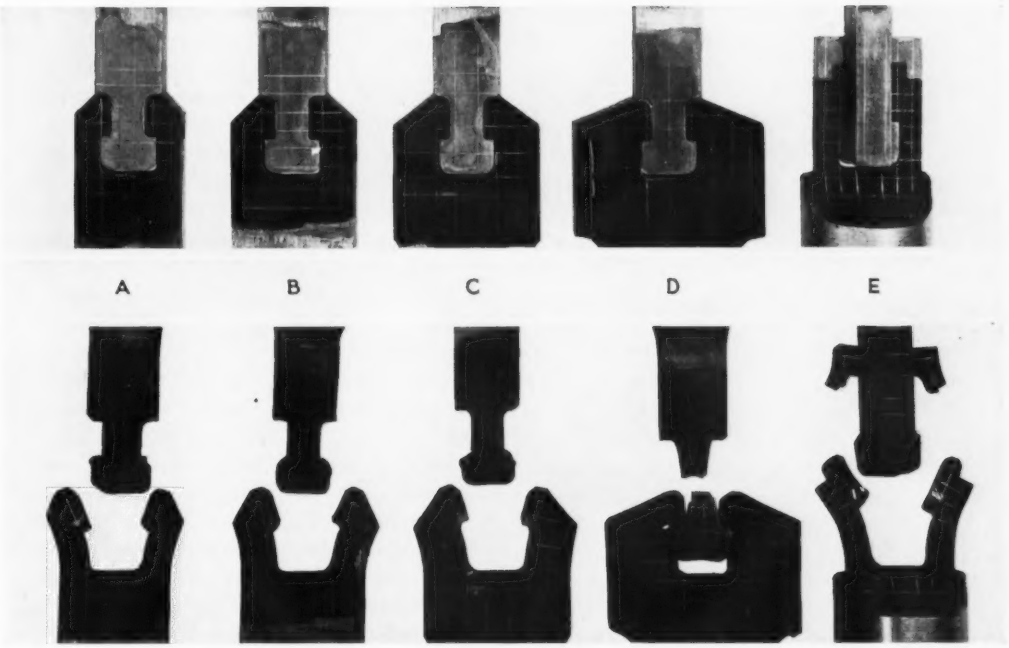


Fig. 17.—Destruction tests carried out on blade fixings of various proportions.

since, in the narrow sections of metal considered, the necessarily considerable changes in area which occur abruptly, lead to a localised concentration of stress which is difficult to predict with accuracy.

Specimens A to D, inclusive, represent standard inverted " T " blade roots in discs having jaws of different thickness ; specimen E represents a special type of blade fixing, having a side grip (from which it takes its name). As will be seen from the following tabulated failing loads of the five specimens, the provision of the side grip adds very considerably to the strength of the attachment as compared with the normal type of attachment of equivalent jaw width represented by sample A. The side grip attachment is used only in special cases of high jaw stresses where the normal method of attachment would be rendered difficult owing to the increase of disc stress resultant upon the use of a heavier jaw.

Specimen	A	B	C	D	E
Load at failure (tons)	1.6	2.65	3.15	4.15	3.0

The problem of blending the change from one section of metal to another by gradual means in order to minimise the effects of stress concentration at points of discontinuity is one which is of extreme importance, particularly where there is the possibility of vibration imposing even more arduous conditions than exist under static conditions, and has received considerable attention in the Research laboratory of the Company's Turbine Department. It follows that the parts demanding the closest attention in this direction are those associated with the rotating members, rotor discs and blading being the items of greatest importance. In blading there are of necessity a number of radical changes in section, which, by virtue of the function they fulfil, must occur somewhat abruptly. First, the change from the blade section to root involves a sudden change from a very thin section to one of comparatively heavy proportion followed by two further changes of section to form the inverted " T " attachment.

In all of these every care is taken to blend the change from one section to the other by radii as large as is consistent with the retention of the

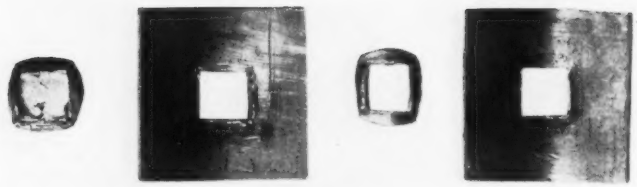


Fig. 18.—Destruction tests of shrouding rivets.

general conformation to fulfil its designed purpose efficiently.

The most drastic of these changes is that occurring at the bottom of the blade proper. At this point the blade edges are called upon to take their proportion of the centrifugal stresses acting over this area, and, in addition, are subject to the bending moment due to the force of steam acting on the blade face. As a consequence there is a tendency towards tearing at the bottom edges of the blade. Cracks, flaws or any inequalities at these points form a potential source of trouble, and for this reason a considerable amount of attention has been devoted to the blending of the blade section into the root and to its accurate and uniform production in manufacture. It has been found that very small tool marks almost in the nature of scratches influence the load at which failure occurs and effectively illustrates the importance of the care which the Company attaches to the manufacture of its blading.

Apart from such tests as a verification of or extension to mathematical analysis, there are many cases where there is no basis whatever for calculable design, and while "judgment" might well be considered as satisfactory in determining the issue, the English Electric Company makes it its definite policy to back up and explore all such rule of thumb methods by positive tests to destruction before adopting the principle to practical design.

Instances of the application of such tests which may be cited include the riveting of shrouding to the tips of blades. Too short a tang obviously would result in a rivet head of insufficient strength to resist the forces tending to lift the shrouding off its blade, while too long a tang would result in the possibility of a loose fixture due to the inability to spread the head of the rivet effectively or else the production of a split head with the added danger of fatiguing the metal by excessive hammering. When considering the adoption of a new blade material for a specific purpose, amongst other things, its suitability for riveting must be considered where shrouding is necessary, and then the length and cross-section of riveting tang must be determined in accordance with the characteristics of the metal and the forces acting. Fig. 18 shows the results of a test on an experimental shrouding rivet.

Illustrative of previous comments on the attention paid to the avoidance of changes in conformation liable to give local points of high stress concentration is the blending of the tang with the top of the blade by means of radii at the shoulders, necessitating a countersinking of the underside of the rivet holes in the shrouding.

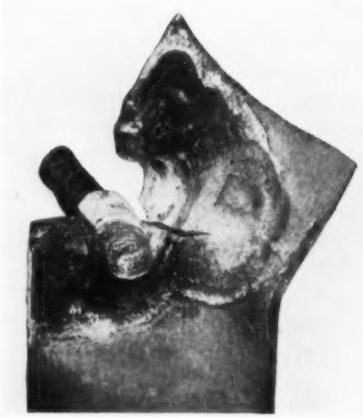


Fig. 19.—Result of test to investigate adherence of lace wire brazing.

Fig. 19 shows the result of a test taken to determine the effectiveness of brazing lacing wires into turbine blades. In this case a sawcut was made from the edge of the blade to the circumference of the lacing wire and the brazed joint was torn apart to show the nature of the braze. Such a

test is necessitated when any change in blading material is proposed where lacing wires are necessary.

Another example of tests taken in the absence of any calculable data for design purposes is that of the determination of the optimum thickness of white metal for thrust pads in a Michell bearing. Several considerations set limits to the thickness of the white metal bearing pads, but within these limits there exists a variety of claims for tendencies to either extreme. In determining the thickness of pad to be adopted a number of experiments were made on various depths of white metal by applying shear and cleavage forces to observe the relative strengths of attachment of the white metal pad to its brass block, and, in addition, the effect of dovetail attachment was investigated.

In addition to mechanical tests of this nature, which might be more properly classified as bearing on the strength of materials, tests on completed units of auxiliary plant are frequently carried out to ascertain their behaviour and to check calculated performance figures, which in many cases cannot be separated for analysis when the plant is running as a whole.

Into this category can be placed an important series of tests carried out on the Company's standard type of main oil pump for steam turbines. This pump is mechanically driven from the turbine shaft.

For the purpose of these tests various standard sizes of this type of pump were selected and coupled in turn to an electric motor providing the driving effort. A pumping load was provided by throttling the oil discharge pipe and the oil quantity was measured by means of an orifice in this pipeline. One series of tests dealt with the distribution of pressure over the pump casing, and as an extension to this series an investigation of the effect of various tooth-to-body clearances was made, which established that there is a most economical clearance beyond which there is a marked increase in friction losses. This bears an analogy with experience gained in disc friction experiments, namely, that there is a definite and most economical clearance between a body rotating in a given medium and its enclosing casing. Clearances both above and below will show an increased friction loss. The critical clearance will, in general, depend upon the nature of the medium in which the rotation takes place, and also on the shape and finish of the surfaces of both stationary and moving parts.

Referring back to the subject of the vibration testing of discs, the necessity for a machine capable of testing discs under their normal running conditions is emphasised by the following considerations. Certain types and sizes of discs have, on occasions, to be employed which have their static critical frequencies of vibration below the corresponding product of running speed and nodal diameters. In these circumstances there is the possibility that the rotating critical frequencies, always higher than the static values on account of the stiffening effect of centrifugal forces, may coincide with or approach the danger region of the nodal multiples of the running speed.

It has already been mentioned (see page 204) that there is a relation between the critical frequencies of a rotating disc, its static critical frequencies and the speed of rotation, namely $(\text{rotating frequency})^2 = (\text{static frequency})^2 + K \times (\text{R.P.M.})^2$, but the factor K varies over a wide range even for discs of similar proportions rotating at the same speed, and it becomes essential, therefore, to ascertain the critical rotating frequencies experimentally in order to avoid, by tuning if necessary, the dangerous regions.

An example illustrating the divergence of the value of K in the case of two very similar discs is shown in Fig. 21. The curves show the static critical frequencies for both discs with different nodal diameters together with the corresponding experimentally determined values of their critical rotating frequencies. Comparison of the calculated values of K from the two pairs of curves shows that for a 2-nodal diameter vibration the values are 3.0 and 2.1 repectively for discs A and B. For discs so similar this represents a large disparity, and in the absence of experimental data a correspondingly large error in determining the rotating frequency from static characteristics would therefore arise in assuming either value of K to hold for both discs. Inspection of the curves will show

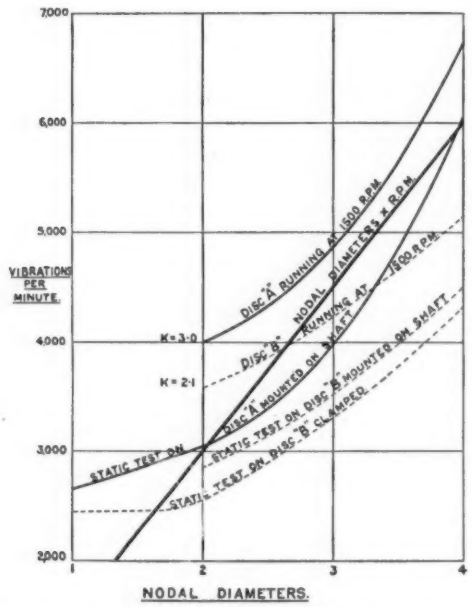


Fig. 20.—Comparison of static and rotational tests of vibration of two turbine discs.

how critical is this value of K as bearing on the determination of safe running, since for certain types of discs such as those considered in Fig. 20 the critical rotating frequencies are unavoidably close to the product of nodal diameters and speed of rotation.

Other factors, such as temperature, method and tightness of fixing of blading, blade loading, disc profile, and method and tightness of fixing of disc to shaft, exercise a modifying influence on the factor K, which still further makes it imperative that the rotational critical frequencies of vibration be determined by actual test under running conditions.

Another interesting comparison illustrating the importance of carrying out these tests under normal operating conditions is afforded by the curves of static tests taken on disc B, first with the disc merely clamped by the hub to a face plate, and second with the disc mounted in its normal designed manner on a shaft.

As a matter of interest it will be noticed that the difference in frequency of vibration at the lower nodal diameters is greater than at the higher nodals. This is in line with previous statements regarding the tuning of discs, that is, alteration of the character of the disc in the region of the hub more particularly affects the frequency of low nodal diameter vibrations, whereas alteration of the rim or of the blade loading has its greatest effect on the high nodal vibrations.

The possession of such a machine as has been described for the vibration testing of discs, besides serving the immediate purpose of testing discs as the need arises, is the means of building up data on which future design is based and progressively reduces the number of tests necessary as experience is gained. The apparatus has a further sphere of usefulness inasmuch that disc windage and friction loss data can be obtained by the use of a torsion dynamometer attached to the driving shaft. Discs up to 100 in. in diameter can be accommodated in the machine for vibration or rotational loss tests.

Apart from mechanical tests of the nature described, the routine testing of materials plays a very important part in the scheme of things. It is a generally acknowledged fact that modern advances in turbine design have been made possible only by the almost revolutionary strides made in metallurgy and the production of special high-tensile and creep-resisting alloy steels. The use of such steels having special properties makes it a factor of prime importance that the designer has every possible assurance that the materials used are completely in accordance with the specification upon which his design is based, and with this object in view scupulous attention is paid to the initial testing of all raw materials employed. Test pieces are taken from every bar destined for blading, and these are tested and the test sheets approved before the bar is liberated to the shops for manufacture to proceed.

The testing and inspection of rotor forgings calls for special care and is necessarily a job for highly skilled and experienced examiners.

Provision is made on all forging drawings for test pieces to be taken from selected points, and from these specimens the grain structure is examined, having due regard to the position from which the piece is taken and to the degree of forging to which it has been subjected. For

instance, a test piece taken from a shaft end, which, as the result of its comparatively small diameter, will have been subject to considerably more forging than the main body of the rotor, will show a more pronounced grain in the axial direction than will a sample taken from the outer periphery. In consequence of this variation of structure, it becomes necessary to apply strength tests with or across the grain according to the direction of the major stress in the finished rotor at the point under consideration. At the same time transverse tests are taken since they give useful information on the homogeneity of the material and show up cleavage planes which may result from impurities or pressed out cavities, etc. Throughout the machining of all rotor bodies a careful watch is kept by skilled examiners for signs of "ghost lines" or other indications of irregularity in the material and a similar examination of the centre of the forging is carried out by means of the borescope illustrated in Fig. 21, which takes the form of a specially developed borescope with its own illuminant for insertion in a hole bored through the axis of the rotor. If any doubts are raised as the outcome of these tests, examination by photo-micrography or by sulphur prints of the structure of the material is made at the point in question.

In addition to the previously mentioned tests which refer to particular cases, chemical, physical, microscopic and mechanical tests are applied systematically to raw materials generally, the value of which lies in their correlated results and in the explanations of divergencies or the verification by chemical or microscopic means of physical or mechanical tests.

For the testing of completed turbine units and for the experimental investigation of steam flow problems, the Works is equipped with high-pressure test boiler plant. This unit, recently installed, was described fully in the May, 1936 (Vol. VIII, No. 2), issue of the English Electric Journal, and was designed to meet the growing tendency towards the use of high pressure and temperatures among all classes of turbine users. The boiler has a normal evaporative capacity of 22,000 lbs. of steam per hour, with a maximum steaming rate of 35,000 lbs. for one hour, which enables turbines to be tested on loads up to about 4,000 kW., depending upon the conditions of test, and up to a pressure of 485 lbs. per square inch and a total steam temperature of 850 deg. F.

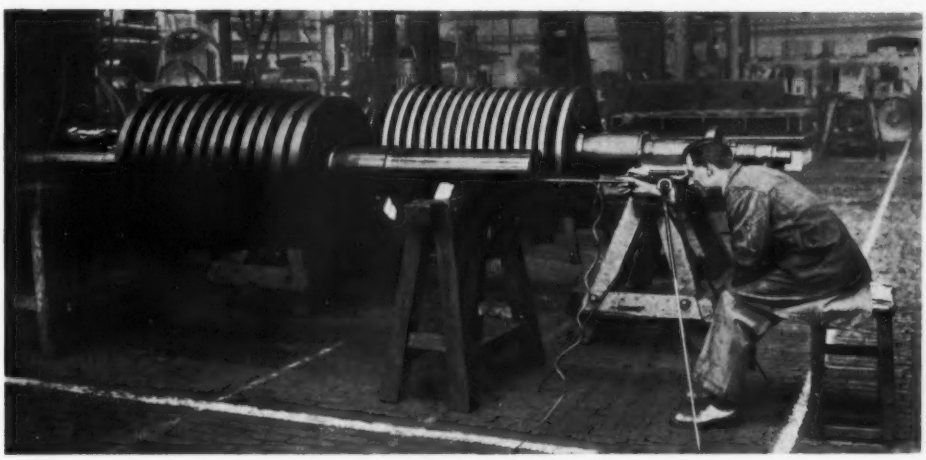
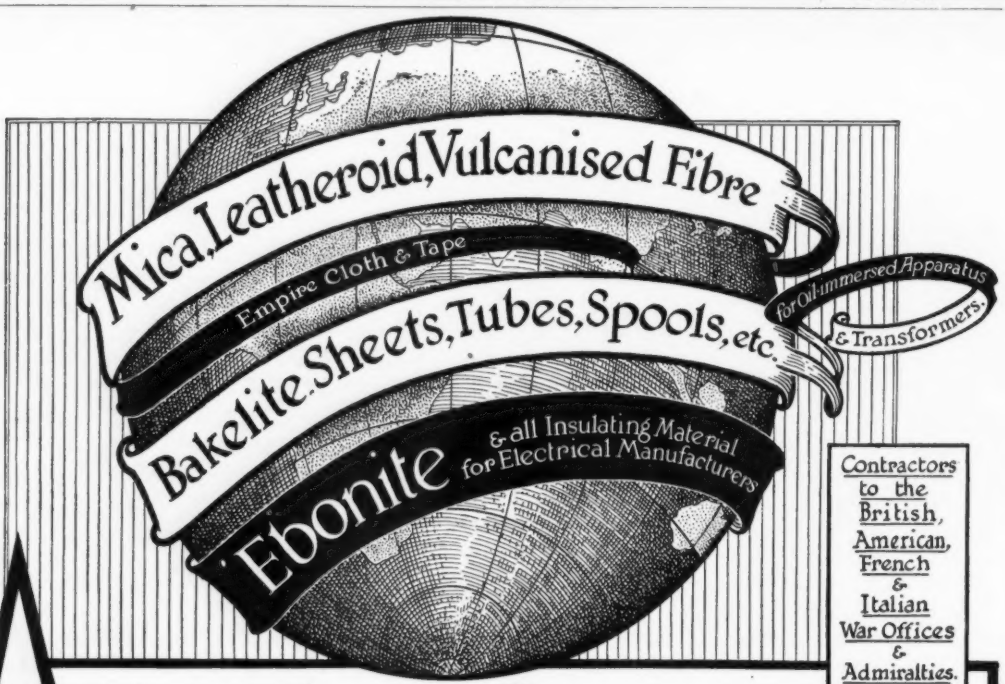


Fig. 21.—Examination by means of the borescope of the internal surface of the hole bored through a turbine rotor.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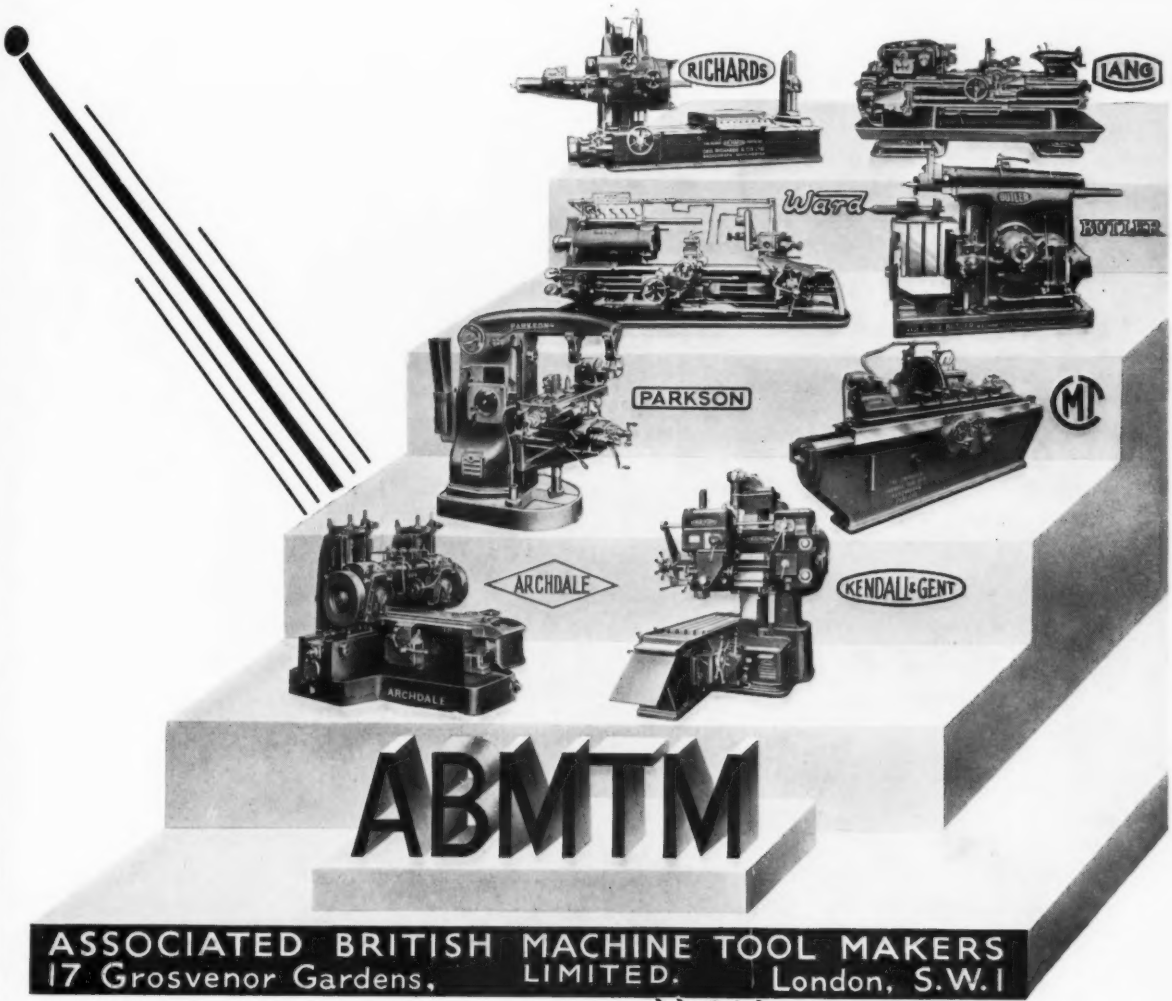

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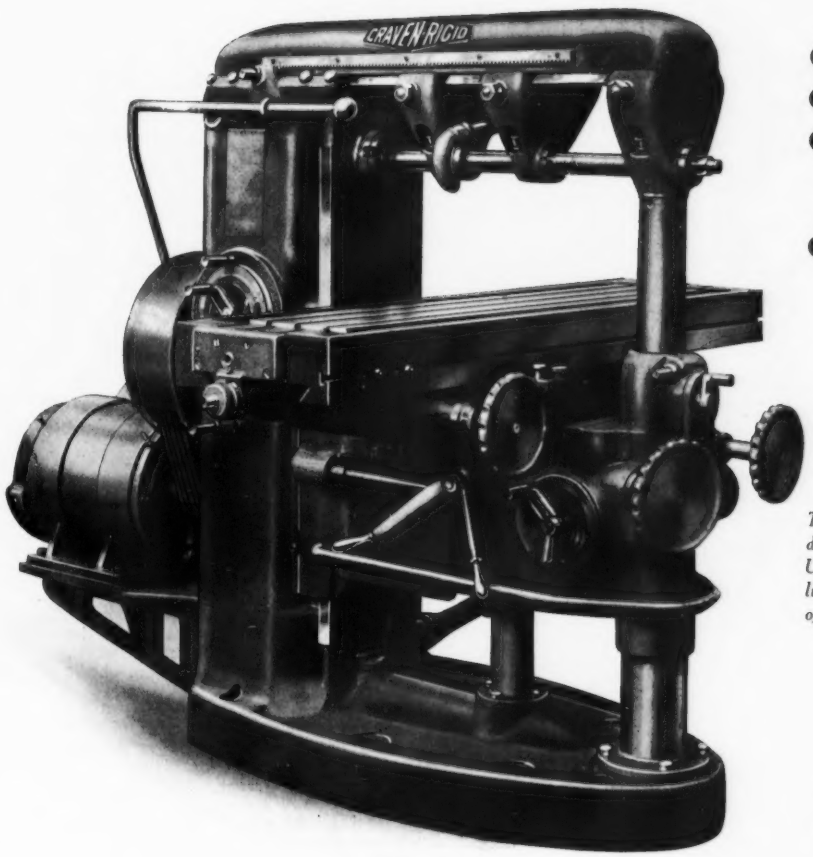

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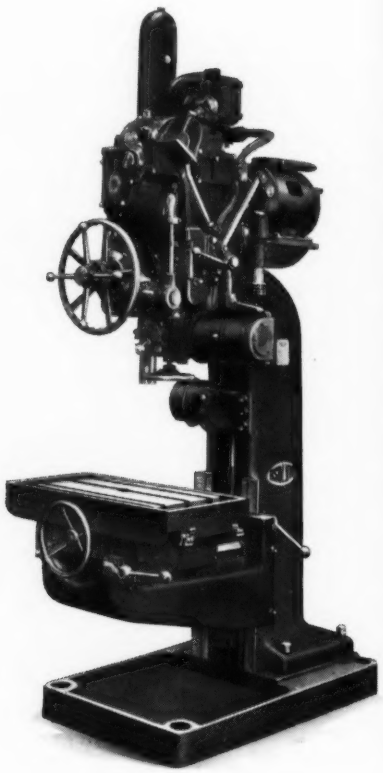

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Marelli & Co. Ltd. .... —
Mason, E. N., & Sons, Ltd..... VI
Micanite & Insulators Co., Ltd. VIII
Midland Steel Co., Ltd. XI
Siemens Brothers & Co., Ltd. X
Silvertown Lubricants Ltd. XIII
Sterling Varnish Co., Ltd. V
Straker Brothers Ltd. ... XVII
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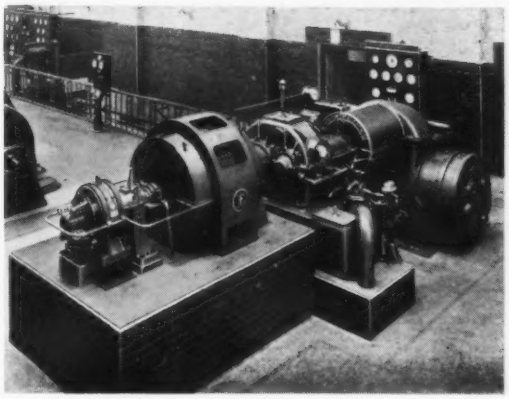

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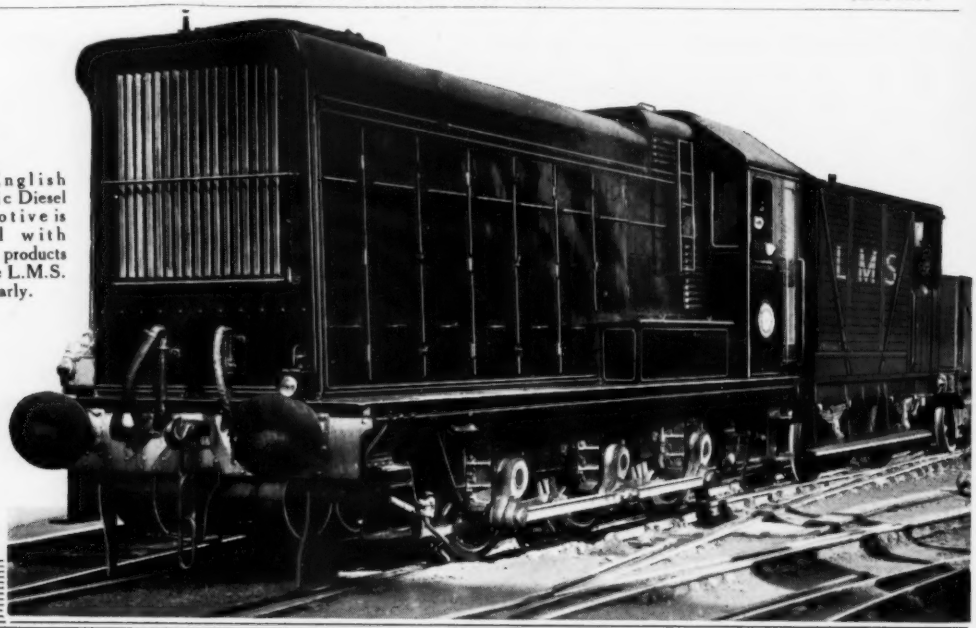

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